**Ship
Structure
Committee**

SR-1266

1981

The main propulsion machinery and shafting aboard
ships has always required foundations which will limit move-
ment. As hulls have become more flexible and horsepower
has increased, the need for rational foundation design
which will link these flexible hulls with the more rigid
machinery and shafting has increased.

The Ship Structure Committee undertook this effort
to study the criteria which the designer may use to adequately
address the problems of meeting distortion limits imposed by
machinery manufacturers due to bearing loading, misalignment,
gear tooth wear, and excessive vibration.

This report presents a proposed methodology for
dealing with these problems and gives an example application.

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Chairman, Ship Structure Committee

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16. Abstract   Recent trends in increased ship hull flexibility, particularly in large ships, have given urgency to a host of problems which were not encountered before in naval architecture. This report deals with one of these problems, specifically the compatibility between local hull deflections and distortion limits imposed by the operational requirements of the main propulsion machinery components. The need to conduct this study was felt because very often problems of shaft misalignment, gear wear, excessive vibration and others, were found to be most probably a result of insufficient stiffness in machinery support systems. The evaluation of the relationship between manufacturer's requirements and the structural design of machinery foundations is the goal of this research effort.

The overall objective of this paper is to derive a set of recommendations capable of helping the designer meet the requirements of foundation stiffness and which are necessary for the good performance of machinery components. The design recommendations to be derived essentially concern the structural arrangement of machinery spaces and support systems. Also included are a group of suggested methods and techniques of structural analysis and design which can assist the designer in implementing these recommendations. As a result, it is hoped that the gap between hull flexibility requirements and machinery operational requirements for a ship can be reduced, so that overall design process and the ship's performance can be improved.

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METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in²	square inches	6.5	square centimeters	cm²
ft²	square feet	0.09	square meters	m²
yd²	square yards	0.8	square meters	m²
mi²	square miles	2.6	square kilometers	km²
	acres	0.4	hectares	ha
	MASS (weight)			
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
	VOLUME			
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft³	cubic feet	0.03	cubic meters	m³
yd³	cubic yards	0.76	cubic meters	m³
	TEMPERATURE (exact)			
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm²	square centimeters	0.16	square inches	in²
m²	square meters	1.2	square yards	yd²
km²	square kilometers	0.4	square miles	mi²
ha	hectares (10,000 m²)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m³	cubic meters	35	cubic feet	ft³
m³	cubic meters	1.3	cubic yards	yd³
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

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INTRODUCTION

Recent trends in increased ship hull flexibility, partic-
ularly in large ships, have given urgency to a host of problems
which were not encountered before in naval architecture [1]*.
This study deals with one of these problems, specifically,
the compatibility between local hull deflections and distortion
limits imposed by the operational requirements of the main
propulsion machinery components. The need to conduct this
study was felt because very often problems of shaft misalign-
ment, gear wear, excessive vibration and others, were found to
be most probably a result of insufficient stiffness in machinery
support systems [2-4], and because of insufficient knowledge
of shipboard environment and flexibility by machinery manufac-
turers. (Ship machinery is usually designed by assuming a
concrete foundation). These reasons show clearly the relevance
of evaluating in a comprehensive way the relationship between
manufacturer's requirements and the structural design of
machinery foundations.

In view of unfortunate past experience, manufacturers
now attempt to scrutinize carefully the environment in which
their equipment must function. In the past, this could be
done by experience and by comparison with similar designs.
While this procedure worked for many years, it became some-
what inadequate as vessel size grew and economic pressures
increased to minimize hull weight and cost. Today more sophis-
ticated methods can be used by the designer to determine
structural response. The proposed solution, therefore, requires:
(a) the machinery designer to specify reasonable limits within
which his equipment can function properly, an area in which
as this study indicates a good degree of agreement has already
been reached by main propulsion machinery manufacturers in
this country, and (b) the hull structural designer to determine
that a support system will meet these limits under all
normal operating conditions.

In the case of ships built in the U.S., hull-machinery
compatibility problems such as those mentioned above have been
found to be relevant in large geared-turbine powered ships with
units in the size range from approximately 25,000 SHP to 50,000
SHP. In fact, most of the design experience in this country in
the case of large ships has traditionally been concerned with
turbine-powered vessels. On the other hand, in Europe and
Japan, diesel engines have often been used for the propulsion
of large ships, and, in Europe, studies on hull-machinery
compatibility have also been conducted on diesel-powered ships
[5-7]. Because of the current world energy crisis, a growing

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* Square brackets designate references listed before the
Appendices.

interest in diesel propulsion is now being felt, and this trend is expected to affect the shipbuilding industry in this country [8-9]. For this reason, the study conducted here also addresses the hull-machinery compatibility problem as it relates to diesel-powered ships. However, the main thrust of this research is concerned with turbine-powered ships. The conclusions and proposed design method can apply to steam as well as gas turbines.

This research program was subdivided for convenience into four main tasks, which followed an extensive computer-aided literature search using the NASIC* Search Service available through the M.I.T. Libraries.

The first task included a survey of major U.S. and foreign machinery manufacturers in order to determine their requirements for rigidity of the main engine supports. Based on this information, a set of general requirements defining maximum foundation deflections, and representing what was felt to be an acceptable industry-wide practice have been defined.

The second task consisted of a review of the design of main engine, gear and thrust-bearing support structures of selected ships, in order to define as much as possible current design practices. This included a study of overall arrangement and scantlings of main support members of machinery, reduction gears, thrust bearing, shaft bearings, and also the dimensions and arrangement of shafting.

The third task was essentially a critical review of available analytical and numerical procedures for studying the coupled response of hull and machinery. Based on this review, it was possible to identify the methods of structural analysis best suited for the study of hull-machinery-compatibility related problems.

Finally, the fourth and last major task was aimed at identifying criteria for defining the structural rigidity of machinery-support systems. This includes recommendations concerning the structural design of these support systems, so that machinery requirements are met, and the possibility of failures due to excessive flexibility is minimized.

The overall objective of this project is to derive a set of recommendations capable of helping the designer meet the requirements on foundation stiffness necessary for the

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* Northeast Academic Science Information Center. The following data bases were accessed by the searchers: MRIS (Maritime Research Information Service) and COMPENDIX (Engineering Index).

good performance of machinery components. The design recommendations to be derived essentially concern the structural arrangement of machinery-support systems and spaces. Also included are a group of suggested methods and techniques of structural analysis and design which can assist the designer in implementing these recommendations. As a result, it is hoped that the gap between strength requirements and machinery operational requirements for a ship can be reduced, so that the overall design process and the ship's performance can be improved.

It can be concluded from the brief overview given above that this project, due to its practical implications involved a considerable information-gathering effort. It included, in addition to the extensive literature survey mentioned earlier, exchange of information with Classification Societies, engine manufacturers, shipyards and shipowners, not only in the U.S. but also abroad. A total of twenty-eight shipyards (twelve in this country, three in Canada, six in Europe and eight in Japan), and nine shipowners (six in the U.S. and three abroad) were contacted. Information was received for twenty-three ships, including fourteen tankers, three LNG carriers, three bulk carriers, one roll-on/roll-off, one container ship and one LASH. The wide cooperation received in the information-gathering effort was an important factor for the successful completion of the proposed work, and the authors are grateful to all those who contributed to this effort.

This report is organized in the following way: Chapter I contains a discussion on the hull/machinery rigidity compatibility problem, including some comments on the causes and effects of excessive hull flexibility, a brief description of the factors which can have a stronger influence on the problem under consideration here, and a review of the various solutions offered in the literature. A case study also is presented, involving a LASH vessel for which considerable data were available.

Chapter II deals with the problem of foundation design. The most relevant structural design parameters are identified, a review of current practice is summarized and some design recommendations are given.

Chapter III presents the result of the survey of machinery manufacturers.

Chapter IV describes a design method proposed by the authors. An example of application is included, involving a 188,500 DWT tanker.

Chapter V contains the main conclusions and gives some recommendations for future work.

CHAPTER I.   THE PROBLEM OF HULL-MACHINERY RIGIDITY COMPATIBILITY

1.   Strength vs Flexibility

In ship structural design, the most widely used measure
of adequacy has traditionally been stress.   The strength re-
quirement insures that the stresses never exceed certain
prescribed levels, so that the structural integrity is not
affected.   It is well known that the criterion for hull
primary bending strength is section modulus.   In reality,
the strength criterion cannot be simply stated in terms of
section modulus alone, since shear stresses can also be
relevant, particularly in the vicinity of the ship's quarter
points.   Besides, the hull girder is subjected to other forms
of loading, such as horizontal and transverse bending and
torsion, and in addition to these primary or overall hull
response forms, secondary and tertiary effects also have to
be considered [10].   In any event, the measure of adequacy
can, in general, be expressed in terms of stress or a combina-
tion of stresses, and since, at present, various methods of
structural analysis can lead to a good estimate of the
stresses in a structure, the designer can be reasonably sure
of meeting the required strength.

In addition to a strength requirement, a stiffness
requirement can also be defined.   This implies that the
structure must be designed to avoid excessive deformations
or deflections which would change excessively the geometry and
prevent the structure from withstanding the prescribed loads.
In the case of bending stiffness, the stiffness (or flexibility)
criterion is obviously moment of inertia, I, since under a
given bending moment, curvature is inversely proportional
to I.   In the case of shear stiffness, the criterion is not
so easily defined, since shear deformations can be a rather
complex function of the cross-sectional geometry, the shear
modulus and Poisson's ratio [11].   In any case, it can easily
be shown that stiffness and strength do not necessarily come
together, which means that for a given general geometrical
configuration the scantlings which lead to maximum strength
are not those which imply maximum stiffness.   Thus, a com-
promise between these two objectives is usually necessary [12].

While in the case of strength, relatively simple material
tests can lead to clear practical design limits, in the case
of stiffness the same is not true.   Upper or lower limits on
allowable stiffness are not easy to define, even in the most
simple structural arrangements, unless very specific operational
requirements are to be met.   The fact that hull stiffness
cannot in practice be changed substantially after the ship is
built is another factor which makes the whole problem of
required stiffness an important one.

2. Causes and Effects of Excessive Hull Flexibility

As mentioned in [1], the interest in fundamental hull girder stiffness was increased when proposals for building vessels entirely of aluminum were first studied [13]. This is obviously a matter of special relevance in the case of deadweight carriers, where weight saving is a particularly important consideration.

Several factors have caused the recent trend in de-creased hull girder stiffness. The most important are [12]:

   i.   *Increased length.*

  ii.   Use of high-strength steels.

 iii.   Less stringent corrosion or wastage allowances.

  iv.   Increased knowledge about structural response, encouraging the use of smaller factors of safety and smaller scantlings.

   v.   Wider use of design optimization techniques, in particular weight minimization, leading also to smaller scantlings.

  vi.   Use of aluminum for superstructure construction.

As a result of increased hull flexibility or limberness, various detrimental effects can take place, affecting the ship's performance to varying degrees of severity. These can best be defined, as proposed in [14], depending on whether their major impact is of a dynamic or static nature, as follows:

Dynamic

a.   Personnel discomfort from propeller-induced or other steady-state vibration and noise.

b.   Malfunction of electronic or mechanical equipment, including main shafting, bearing and gear failures from vibration or excessive displacement.

c.   Unacceptable high-frequency stress peaks in primary hull structure due to impact loads such as slamming.

d.   Fatigue of primary hull structure from the steady-state vibratory response of springing.

Static

e.   Excessive curvature causing premature structural instability failure in the primary hull structure.

f. Excessive deformation when loaded resulting in reduced
    payload capacity in the sagging condition, or lower
    bottom clearance.

g. Excessive hull deformation imposing structural loads
    on non-structural items or components, such as joiner
    bulkheads, piping, propulsion safting, hatch covers, etc.

h. Second-order effects introducing inaccuracies into
    many of the customary naval architecture calculations.

Some of the aspects listed above have already been the
subject of various investigations. In particular, the effects
of decreased hull stiffness upon dynamic response from slamming
and propeller-induced vibration have been studied in [15], the
effects on the whipping bending stress components from slamming,
or fatigue from springing, have been considered in [1], and
the problem of shipboard vibration and noise control is
reviewed in [16].

In the present study, the problem of hull-machinery
foundation rigidity compatibility will be studied from a
strictly static point of view, so that it essentially falls
under (g) above. It is obvious that dynamic effects can also
affect the interaction between the hull and the machinery
foundations, not only because of the dynamic distortions on
the hull caused by ship motions, but also because of the
intrinsic dynamic nature of the machinery components [17,18].
This is a subject which will be addressed in more detail at
a later stage.

3. Factors Affecting the Hull-Machinery Foundation Compatibility

a. Static primary deformation of the ship's hull girder.

This is the primary ship structural response, in which
the ship's hull girder is treated as a simple free-free
Bernouli beam. Wave hogging and sagging conditions are usually
taken into consideration, and the effect of quartering seas
can also be allowed. Normally, the primary concern is
vertical bending, but horizontal and transverse bending can
also be taken into consideration.

In addition to flexural deformations, shear deformations
can also bring an important contribution to the overall hull
girder distortions. Taylor [19] found this contribution to
be as much as 19% of the total hull deflection, so that it
should not be disregarded. The same opinion is expressed in [2].

In [2], it was found that in the case of tankers with machinery aft, the hull girder curvature in the machinery compartment would essentially have an opposite sign as in the remaining part of the ship. Thus, if the ship is in the light condition, the hull would in general deform in hogging while the double bottom in the machinery compartment would deform in sagging. The converse would happen in the fully loaded condition. This indicates how a careful computation of the hull girder deflection can help in detecting the possibility of incompatibility between the hull and the machinery.

b.   Dynamic primary deformation of the ship's hull girder

Vibration effects on the hull girder can obviously affect the compatibility between hull and machinery. The same can be stated with respect to hull bottom impact or slamming [17-19].

c.   Thermal effects

Thermal effects due to oil, seawater and steam can have a considerable impact on the deflections of double-bottom and foundations of turbines, gear and gear casing. These effects are in general taken into account when designing the machinery support systems [2].

d.   Lineshaft alignment and vibrations.

Misalignment and longitudinal, lateral and torsional vibrations induced into the shafting by the propeller and/or the propulsion plant should be considered [2].

e.   Shaft stiffness

Due to larger installed horsepower and a tendency toward single-screw ships, shaft diameters have increased and, as a result, lineshafting stiffness has also substantially increased. Since, on the other hand, the hull stiffness has in general decreased, this fact can also be a source of incompatibility between the hull girder and the machinery foundation [2].

f.   Ship's beam

The structure of the double bottom is usually transversely framed, so that as the beam increases, its flexibility also suffers an increase, which can only be compensated by increasing the scantlings of the double-bottom structure. If this is not achieved, the machinery-foundation stiffness might be too low, and this can obviously lead to possible incompatibility between the hull and the machinery. Note that this beam effect

can be quite relevant, since the deflection is essentially
proportional to the fourth power of the span [2].

g. Local deformations

The double-bottom structure is essentially composed of
stiffened panels supported by floors and side shell. The hull
itself is also an assemblage of stiffened panels supported
by transverse bulkheads and web frames. Hydrostatic pressure
and dead loads act on these panels and produce local deforma-
tions which can also affect the hull-machinery compatibility.
Local deformations and insufficient double-bottom stiffness
are in part responsible for the motions of rocking and tilting
of the thrust block, known to have a very detrimental effect
on reduction gears and bearings [2]. These motions are
amplified by the fact that the thrust block can be considered
as a cantilever beam embedded into the double-bottom struc-
ture with an overhung load. This cantilever effect is obviously
more pronounced for larger spans, i.e. when the thrust is
applied at a greater height from the double bottom, a factor
which should carefully be weighed in designing the machinery
layout.

h. After body shape

The after body hull shape can have an important impact
on the local hydrostatic pressure loading on the hull, and
this can also affect the hull-machinery foundation compatibility
problem particularly if the machinery spaces are aft. If
the stern is full or spoon-shaped, the hydrostatic pressure
forces on the side shell are likely to be more important
than the corresponding forces on the bottom. In the case
of a transom type stern, the opposite is in general true.
Thus, the two extreme hull after-body shapes affect differently
the overall and local loading on the ship, in the sense that
while one normally implies excessive buoyancy on the hull
girder and large pressures on the shell plating aft, the other
does not.

The after-body shape can have another important impact
on the hull-machinery compatibility problem by the way it
influences the machinery spaces general shape if located aft.
In the case of a tanker, for example, as represented schemati-
cally in Fig. 1a, the machinery space can be quite narrow in
way of the reduction gear casing. The short floor span is
very stiff and can normally provide adequate machinery support.
In other ships, such as the LASH (discussed in detail in
Section 5) the machinery space is essentially square (Fig. 1b).
In way of the reduction gears, the floor span is very large and
the stiffness is greatly decreased, particularly if the reduc-
tion gear is not close to a transverse bulkhead. This factor
is obviously related to the beam effect discussed in (f) above.

AFTER BODY SHAPES
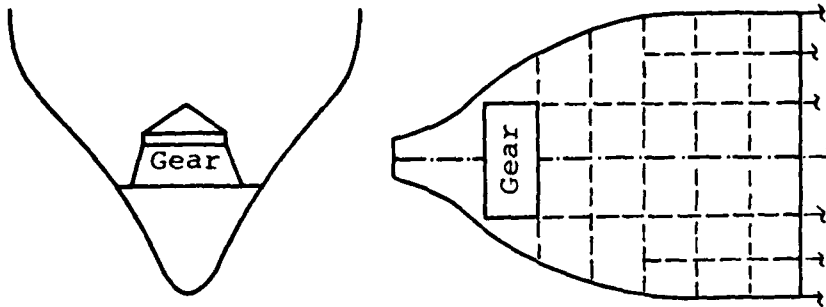
(a)
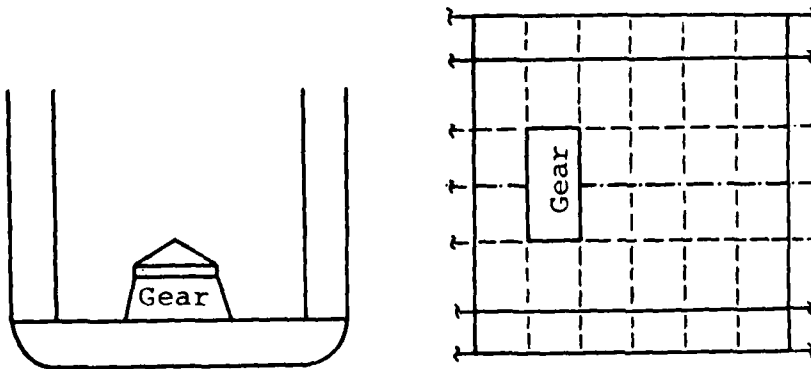
(b)

Figure 1

i. Machinery characteristics

The machinery type, size and location can also be expected
to affect the compatibility between hull and machinery. Larger
units produce larger concentrated loads at the supporting points,
so that the foundation stiffness becomes critical. The
machinery location along the hull is also an important con-
sideration, since the hull stiffness is not constant through-
out the ship's length. The shafting length and number of
bearings are also important parameters, since they affect
directly its stiffness.

j. Draft changes

Local hydrostatic loading on the hull is obviously directly
related to the ship's draft. If large draft changes can
occur between the fully loaded and light conditions, such
as normally happens in the case of tankers, then the local
hull deformations can also vary largely, and this can also
affect the hull-machinery compatibility problem.

4. <u>Brief Review of the Solutions Proposed in the Literature</u>

In order to reduce the possibility of hull-machinery
incompatibility, various solutions have been proposed in
the literature. Essentially these can be classified under
three main categories as follows:

(a) reduce the stiffness of the shafting, adjusting the
     equipment to the increased flexibility of the structure;

(b) increase the stiffness of the foundation and double-
     bottom structure, and, thereby, adapt the structure
     to the support requirements of the machinery;

(c) modify the design of machinery components so as
     to adapt them to the increased flexibility of the
     hull.

In the first group, (a), we can include various possible
alternatives, such as the curved alignment of the line shafting.
In fact, it is well known that the straight alignment of
shafting does not provide a proper operation of the main gears,
which leads to the necessity of a rational curved alignment [20].
In addition, factors such as a careful choice of the number of
bearings, the rational positioning of the first bearing aft
of the main gear with respect to the main gear or diesel engine
and the position of the thrust bearing, must also be con-
sidered [21]. This subject will be discussed in the next
section when describing the modifications introduced in the
original LASH design. This case study along with the example
described in Chapter IV fully outline the steps the designer
should take in situations such as this one.

In the second group, (b) several possible alternatives for increasing the foundation stiffness have been proposed such as reinforcing the thrust-bearing foundation and the adjustment of the web frame thickness [21]. The next Chapter, discusses in detail the subject of foundation design, which is of major importance in problems of this nature.

In the third group, (c), several solutions discussed in the literature can be included. One relates to a new type of bull-gear design termed a "transflex bull gear" [23]. The novel feature in this design is a flexible diaphragm plate which transmits to the gear wheel rim less than 1/30 of those forces and couples transmitted by conventional design. Thus, if this new design is adopted, some of the problems related to hull-machinery compatibility could be reduced.

Another possibility suggested in [2] is the introduction of a flexible coupling between the main gear shaft and the intermediate shaft.

Still another possible solution deals with diesel engines. A box girder design of the machinery base between bedplate and cylinder block, rather than a design based on columns is known to increase substantially the rigidity of the combined engine-hull structure foundation [8,23]. As a result, the double-bottom distortions are reduced by the engine itself with a considerable margin of safety, reducing the possibility of hull-machinery incompatibility. In the case of medium-speed diesel engines, improved designs for reduction gears have also been proposed, with the objective of reducing the detrimental effects caused by excessive hull and machinery flexibility [24]. The subject of diesel engines is considered in detail in Chapter III.

5. A Case Study: The LASH Vessel

5.1 Introduction

In late 1970, the first of twenty large barge/container ships of the LASH type was delivered to its owner after successful trials; however, in the next two years half of these vessels developed machinery troubles that were found to be caused by an incompatibility between the flexibility of the hull structure and the degree of rigidity required for proper support of the machinery. This costly experience, together with problems of a similar nature encountered by some large European vessels, led to recognition of the need for a better and wider understanding of hull/machinery compatibility. The account of the difficulties with the LASH vessels which follows is based on the condensation of a very large amount of test data, experience, and analysis and is not intended to represent a detailed history.

5.2 First Group of Ships

The machinery arrangement, machinery foundations, and hull structure aft of amidships were essentially the same for each of the first eleven LASH vessels. The main particulars of the LASH vessels are summarized in Table I.

The main propulsion machinery consisted of a 32,000 SHP steam turbine driving a single propeller through a standard locked-train, double-reduction gear. All vessels of this first group experienced distress on the reduction gear teeth in varying degrees of severity during their early service life and replacements for several gears were required.

5.3 Second Group of Ships

Modifications made to the machinery arrangement, main shafting, and hull structure of nine vessels comprising the second group eliminated the gear problems. Generally, these changes were retrofitted to the first group and now both groups have operated successfully for many years.

5.4 Gear Distress

Operation of the main machinery in the first three ships was apparently satisfactory when delivered. Following trials of the fourth ship in mid-1971, however, inspection revealed evidence of distress on the second-reduction gear teeth with heavy loading at the forward ends of both helices. Pitting and scuffing led to rapid deterioration and eventual replacement. Subsequent examination of the first three vessels indicated similar distress although very much less severe; a pattern that generally was repeated in the remaining vessels of the first group. There were no signs of distress in the first-reduction gears.

Initially, the reasons for the gear problem were not understood. Attention was focused on the internals of the gear with a detailed analysis of the gear design by the manufacturer, consultants, and shipbuilder. Modifications were made to the gear in those areas that were suspect; however, these internal changes apparently did not eliminate the basic problem and the gears continued to show increasing distress.

Signs of heavy loading on the gear teeth at the forward (or aft) ends of both helices are generally an indication that the gear and pinion axes do not remain parallel during operation. Fig. 2 illustrates how varying amounts of misalignment significantly affect the tooth contact across the mesh.

TABLE I

LASH MAIN CHARACTERISTICS

Principal Dimensions

Length BP	724'
Breadth	100'
Depth	60'
Draft	28'
Displacement	32,650 tons

Machinery

Steam turbine	32,000 SHP

Engine Room Construction

Transverse framing, spacing 7'-4"

Engine room length 73'-4"

Engine room width in way of reduction gear 70'-4"

Web frames at every frame

Tank top - plating thickness = 3/4"

Bottom C.L. girder 3/4" thick

Bottom side girders 9/16" thick

Double bottom depth 8'-9"

Spacing between longitudinals 6' average

Shafting Details

Line shaft diameter 21.88" (original)

Tail shaft diameter 28.56" (original)

Thrust bearing location aft of #2 bearing (original)

Number of line shaft bearings 3

Height of thrust bearing center above inner bottom 7'

LASH 2nd REDUCTION PINION/GEAR MESH
TOOTH CONTACT VS ALIGNMENT
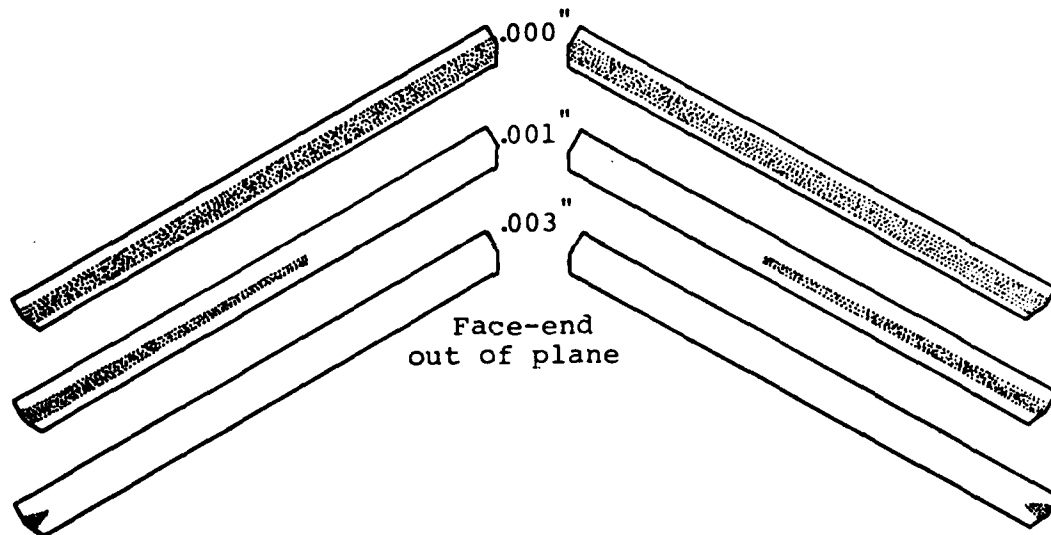
FIGURE 2

The second-reduction, or bull-gear shaft, when connected becomes a part of a continuous beam system supported by a series of bearings. For convenience, these bearings will be numbered from the forward end. It is customary for the gear manufacturer to specify the maximum allowable difference (ΔR) between #1 and #2 bearing static reactions. One manufacturer has based this limit upon a maximum mismatch, or opening between the teeth of meshing pinions and gears, of approximately 0.0002 inches per foot of face width [25]. With approximately five feet between centerlines, this is equivalent to a relative movement of 0.001 inches between #1 and #2 bearings. Generally, ΔR falls between 20-30 per cent of the static reactions [26], and in the case of LASH was established by the manufacturer as 12,400 pounds.

5.5 Bull-Gear Monitoring System

In order to determine what was happening, an electronic system was developed by the manufacturer to continuously monitor the journal position within the oil clearance of each bull-gear bearing. A simplified diagram of the system is shown in Fig. 3. Two proximity probes located in each bearing serve to measure gaps "A" and "B". This enabled the system to display a dot for each journal on an oscilloscope screen, each dot representing the center of the corresponding journal. Electronic magnification permitted movements as small as one half mil to be measured. The display was adjusted initially so that the two dots (forward and aft bearings) were superimposed when both journals were at rest in the bottom centers of their respective bearings. In this position, pinion and gear centerlines were parallel as manufactured and later confirmed by tooth contact tests after installation. Although the journals move to other positions as speeds and loads increase, both journals should move in the same manner if the pinion and gear axes are to remain parallel. Thus, any spread between the dots which develops in operation is a measure of the misalignment of the gear relative to the pinions.

It was found that the bull-gear did in fact skew as power and speed were increased. Accordingly, the position of the first line-shaft bearing was adjusted during operation and the gear could be made to operate in a parallel position at either low power or full power, but no single adjustment would allow proper operation through the entire power range. This suggested that there might be relative movements between the gear bearings and the line-shaft bearings as power was increased.

5.6 Structural Deflection Tests

Test arrangements to measure structural deflections of

AN ELECTRONIC SYSTEM TO MONITOR THE JOURNAL POSITION

WITHIN EACH BULL GEAR BEARING DURING OPERATION
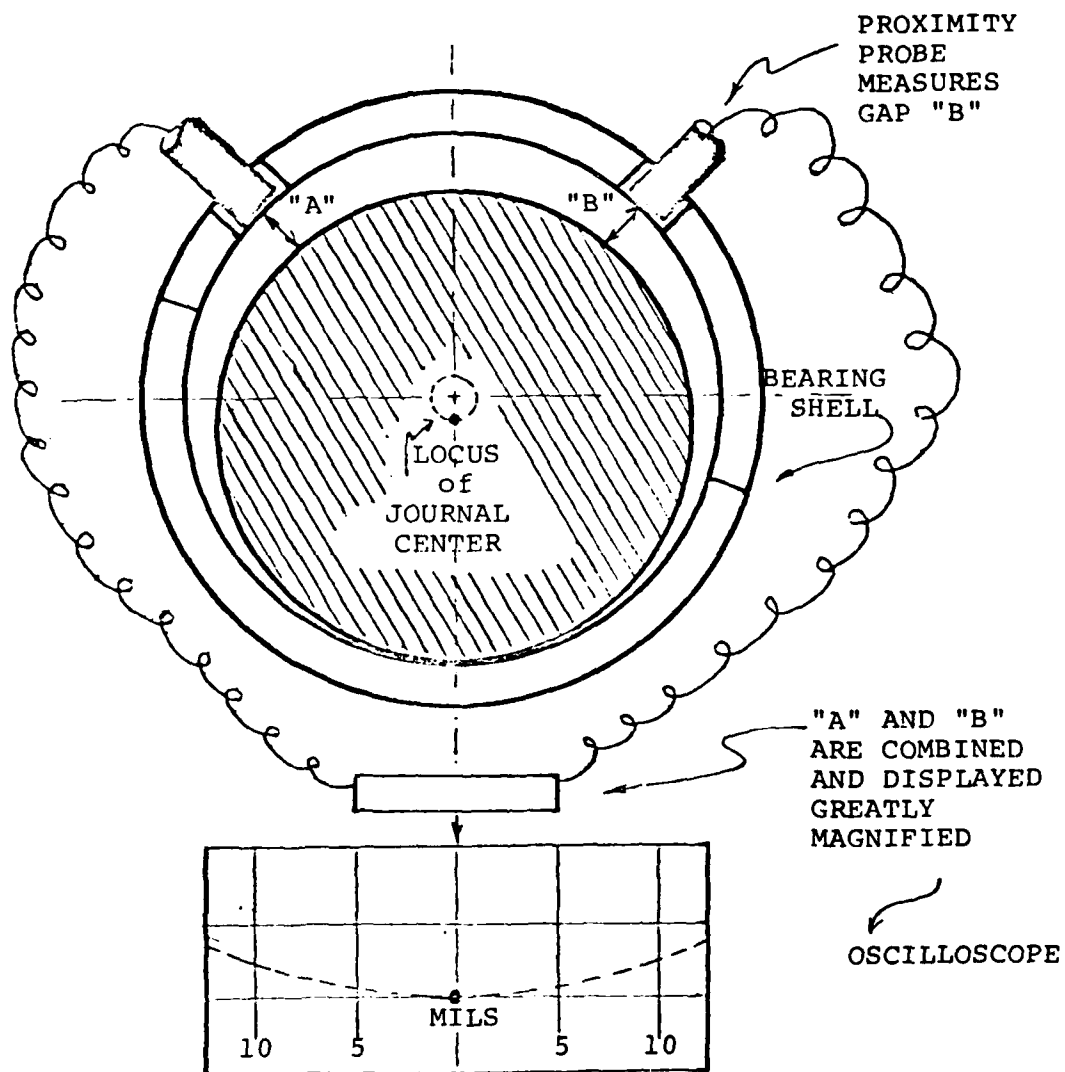
FIGURE 3

gear case and its foundations were developed for the tenth
vessel and data were obtained while underway at full power.
A second series of tests were made on an earlier vessel while
at sea using different test methods and equipment. Finally,
the test arrangements of the tenth vessel were applied to
the eleventh vessel and data were taken at dockside where
full-power torque and thrust were simulated by special
hydraulic devices. The deflections measured in these tests
by different methods showed reasonably good agreement. Data
taken during dockside tests have been chosen for illustrative
purposes because it was possible to apply torque and thrust
independently.

5.7   Torque Test

The direction and magnitude of deflections at selected
points on the gear case and its supporting structure while
under simulated full-power torque alone are shown in Fig. 4.
The forces due to torque reaction are downward on the port side
and upward on the starboard side. The structure supporting
the gear deflects in a corresponding manner and if the athwart-
ship movements are plotted it will be found that each deflec-
tion is approximately proportional to its distance from a
longitudinal axis somewhere in the inner bottom. The entire
gear case, therefore, rotates to port as shown in the exagerated
view of Fig. 5. The movement of the bull-gear bearings rela-
tive to the line-shaft bearings is about ten mils and begins
to explain why a satisfactory alignment could not be established
throughout the power range. The tilt at the foundation is
greater at the aft end by about two to three mils, thus the
gear case is twisted and the pinion axes are skewed relative
to the bull-gear axis.

5.8   Thrust Test

The deflections which were caused by the application
of full-power thrust only are shown in Fig. 6. In this
test, all movements of the gear case were due to deflection
of the foundation because there were no forces or moments
applied to the gear case. The three mil readings at the
lower aft corners were considered invalid and were assumed to
be about six mils in agreement with other data on the aft end
of the gear case. The forward movements of the gear case
were not harmful since they were parallel to the gear and
pinion axes. The five to seven mil depression, however, was
significant since it changed the position of the bull-gear
bearings relative to the lineshaft bearings.

DEFLECTIONS DUE TO TORQUE

Figure 4

GEAR CASE ROTATION

19,000,000 IN.LBS.

PORT   STBD
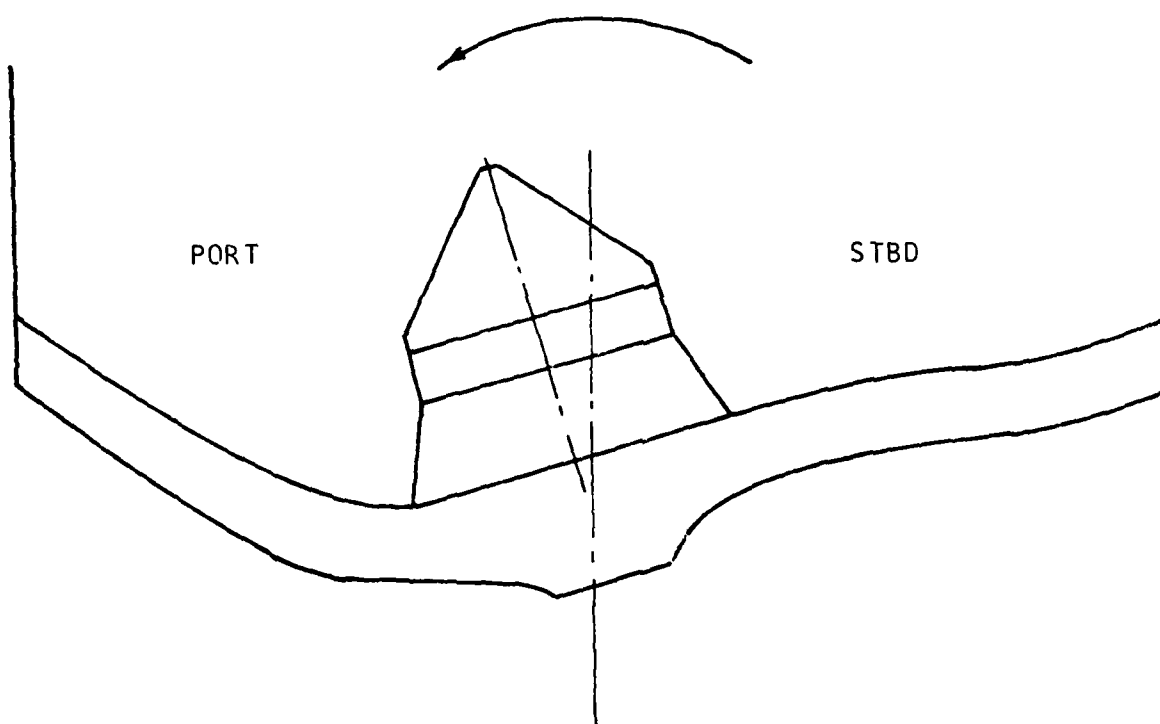
Figure 5

DEFLECTION DUE TO THRUST
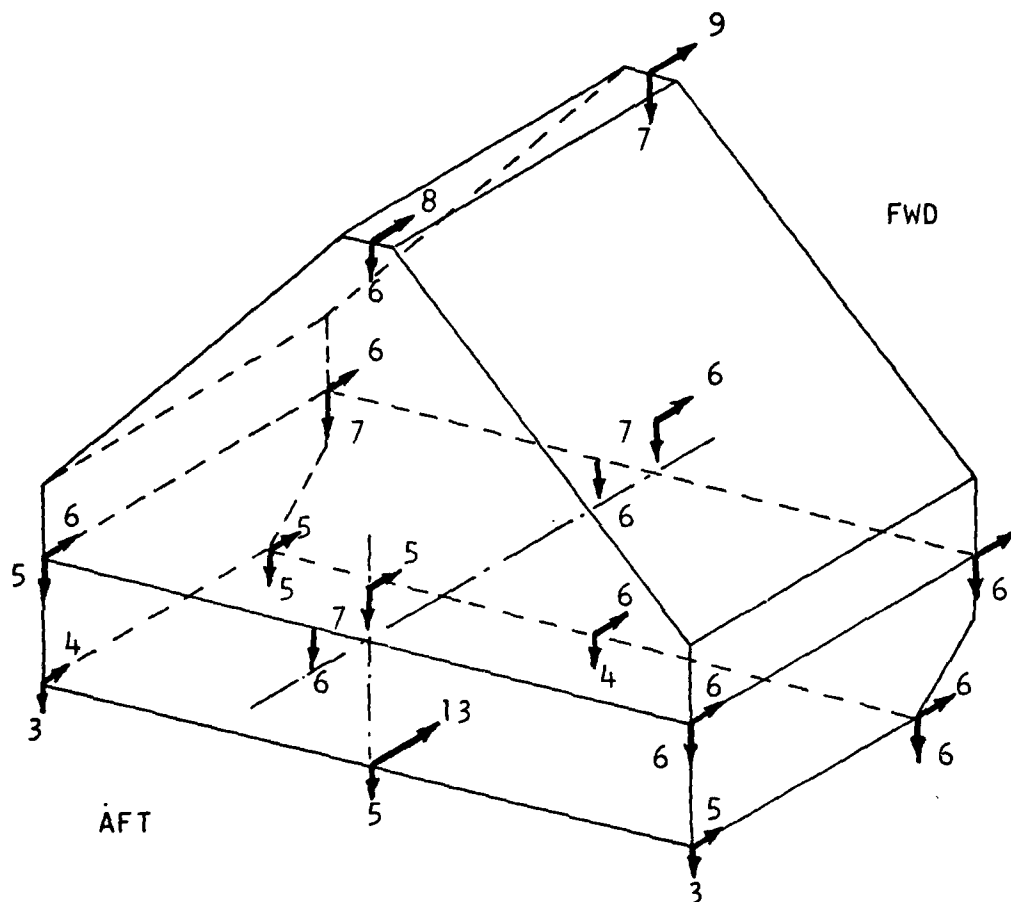
3 MIL VALUES AT LOWER AFT CORNERS ARE INVALID DUE TO POOR
INDICATOR MOUNTING
ALL MOVEMENTS OF THE GEAR CASE ARE DUE TO FOUNDATION
DEFLECTION BECAUSE THERE ARE NO FORCES OR MOMENTS DIRECTLY
APPLIED TO THE GEAR CASE
THE UNIT IS DEPRESSED ABOUT 5-7 MILS

Figure 6

5.9 Combined Torque and Thrust

The measured deflections due to torque and thrust may be combined as shown in Fig. 7. The encircled values were developed from the results obtained by a finite-element analysis program and in most cases are in reasonable agreement with the measured results considering the instrumentation problems and the complexity of the calculations.

5.10 Main Thrust Bearing

The depression of the bull-gear bearings appears to have been caused by the application of thrust at the main-thrust bearing just aft of the gear. Fig. 8 illustrates the arrangement of the thrust bearing and gear foundations and shows that the main-thrust bearing moved forward 20 mils and downward 5 mils. The motion was essentially rotation as shown in Fig. 9 with the lower thrust shoes becoming more heavily loaded. Multi-shoe thrust bearings have devices which are intended to equalize the loads on the shoes, or pads; however, research has shown that these arrangements are not always effective. Reference [28] states "leveling links are unable to follow shifting of the housing alignment with full thrust load, and force gauges show some pads to be taking nearly the entire load." The tests indicated this effect to be present at loads down to twenty per cent of rated thrust. Failure to equalize the loading of the pads was apparently caused by friction at the pad and link contacts and the attempts to release this friction by applving a vibration shaker to the housing were not successful.

An eccentric load at the thrust collar would introduce a bending moment in the shaft which would tend to unload the #2 bearing. Based upon the measured rise of #2 bearing between zero to full thrust of two mils, and by reference to the shaft flexibility characteristics, it has been estimated that ΔR could be changed by as much as 80 per cent of the maximum allowable value, a significant amount.

5.11 Dynamic Deflection Due to Rolling

The instrumentation shown in Fig. 10 was applied to one vessel of the first group of ships to measure relative deflections of the forward and aft sections of the gear foundation while the vessel was at sea. Dynamic deflections in the athwartship direction of 2-1/2 - 5 mils were recorded with the vessel rolling through a total amplitude of 8-13 degrees. Large roll angles, such as occur in heavy weather, were not encountered during the test and no further measurements are available; however, the data appear to indicate that relative

DEFLECTION DUE TO COMBINED TORQUE AND THRUST

FOUNDATION TWIST_____2MILS
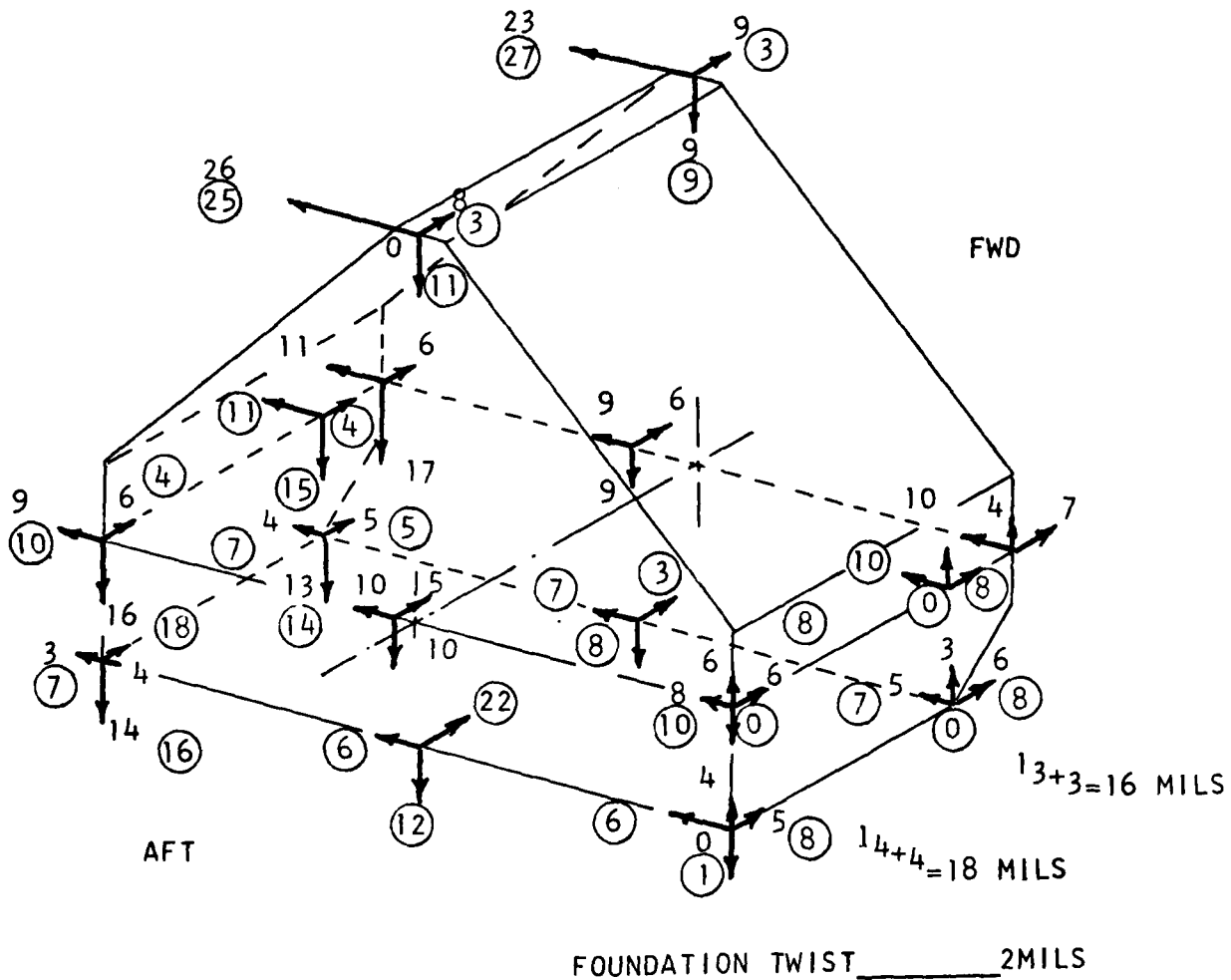
CIRCLED FIGURES ARE CALCULATED BY THE ABS PROGRAM "DAISY"
FIGURES WITHOUT CIRCLES ARE BASED ON MEASURED DEFLECTIONS
ALL VALUES ROUNDED TO THE NEAREST MIL

Figure 7

MEASURED DEFLECTION OF THE THRUST BEARING FOUNDATIONS
DUE TO FULL POWER THRUST
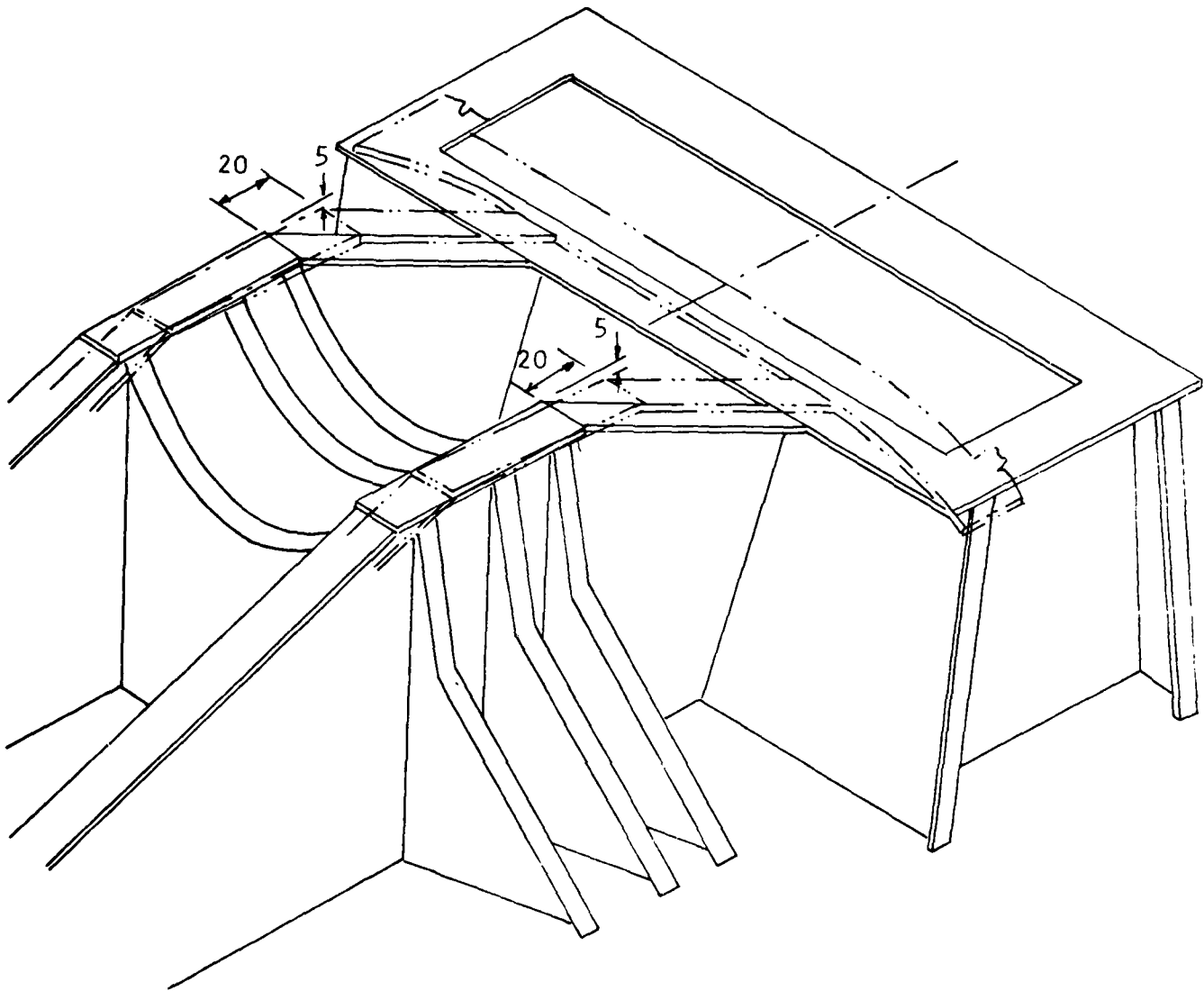
ALL VALUES ARE MILS

Figure 8

MAIN THRUST BEARING
FOUNDATION DEFLECTION

THRUST HOUSING
TILTED BY DEFLECTION
OF
FOUNDATION

.040"

FULL POWER THRUST

.020"

REACTION

ANNULAR SPACE
FOR
THRUST SHOES AND LEVELING LINKS
(NOT SHOWN)
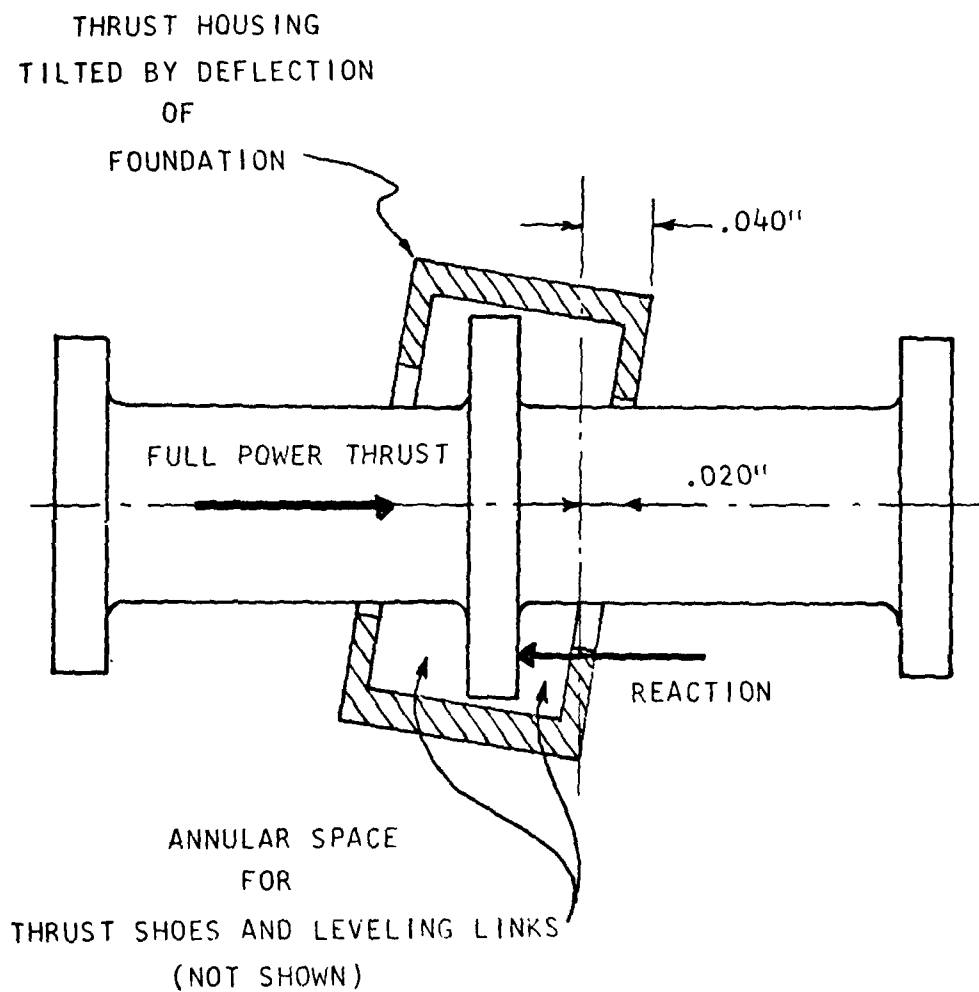
Figure 9

RELATIVE DEFLECTION OF THE FORWARD AND AFT SECTIONS OF THE GEAR
FOUNDATION MEASURED AT SEA WHILE THE VESSEL WAS ROLLING

ROLL ANGLE	RELATIVE DEFLECTION	
DEGREES	MILS	
	V	A
5s-3p = 8	$\frac{1}{2}$	$2\frac{1}{2}-3\frac{1}{2}$
6s-4p = 10		3-4
$7\frac{1}{2}$s-$5\frac{1}{2}$p = 13		4-5
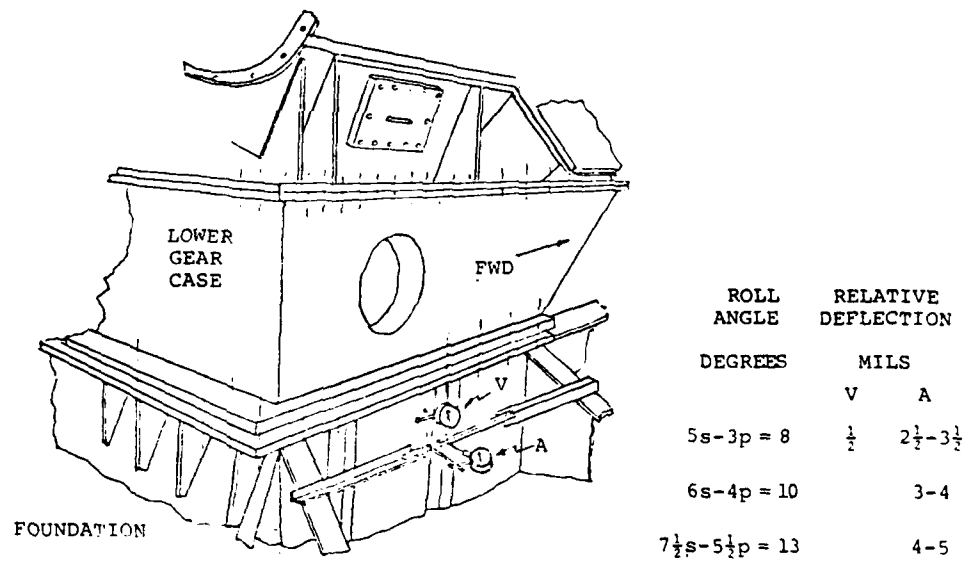
Figure 10

deflection increases with roll angle. This would imply that very significant deflections may occur with large roll angles.

If the unit is assumed to be in alignment under static conditions, a positive/negative nonparallel condition would occur at the gear mesh in each roll cycle and would be expected to cause heavier contact at both ends of each helix. This condition was reported on several vessels, thus tending to support the dynamic deflection measurements.

When small metallic particles are found on the magnets fitted in the lubricating oil strainers, they generally come from deteriorating tooth surfaces. Such particles were often found on those vessels that suffered severe tooth damage. It was noticed that the rate at which particles collected usually increased during heavy-weather conditions. It is also possible, of course, that some of this effect may have been due to the agitation of the lubricating oil sump which stirred up particles that had been settled at some previous time.

5.12 Shafting System Modifications

Three important changes were made to the eleventh vessel: (Fig. 11)

a. The line-shaft diameter between #2 and #3 bearings was reduced to the minimum allowable with the existing material.

b. The #3 bearing was moved aft

c. The main thrust bearing was relocated to a position aft of #3 bearing.

The effect upon shaft flexibility is illustrated in Fig. 12. Calculations indicated the gear case could now undergo equal vertical movements of #1 and #2 bearings (parallel) of $\pm$ 22 mils instead of $12\frac{1}{2}$ mils without exceeding $\Delta R = \pm 12,400$. This method of measuring shaft flexibility has been called "allowable setting error" [29] and should include (a) installation tolerance, (b) hull/foundation deflection, and (c) error in estimating the thermal rise of foundations and gear case. An absolute minimum value of $\pm$ 10 mils is recommended by reference [29]; however, reference [30] lists a number of ships which have operated between $\pm$ 10 mils and $\pm$ 6 mils. Installations with less than $\pm$ 6 mils were generally in difficulty and required modification.

The allowable vertical movement of one gear bearing (nonparallel) is considerably less but increased from $\pm$ 2-1/2 mils to $\pm$ 4 mils. The allowable movement of #3 bearing relative to the gear increased from $\pm$ 10 mils to $\pm$ 14-1/2 mils.

CORRECTIVE MEASURES TO SOFTEN THE SHAFTING SYSTEM

INITIAL DESIGN

#3 LINE-SHAFT BEARING MOVED AFT

MAIN THRUST BEARING MOVED AFT

LINE-SHAFT DIAMETER REDUCED

73'4"
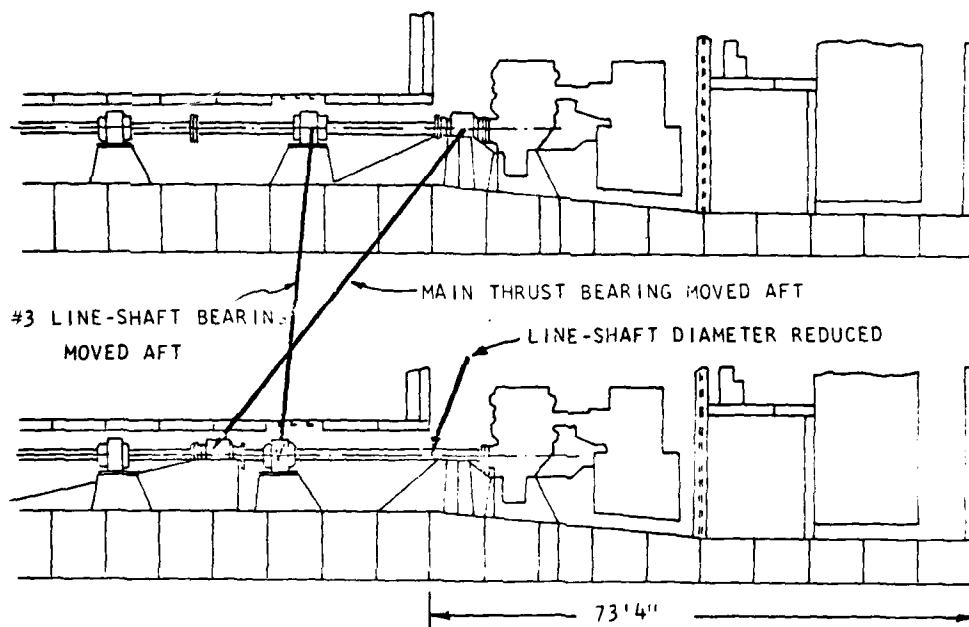
MODIFIED DESIGN

Figure 11

MAIN SHAFTING

INITIAL

4   3   BHD   2   1

FLEXIBILITY BASED UPON
$\Delta R = R_1 - R_2$
$= \pm 12,400$ LBS. MAX.

$23\frac{3}{16}$

THRUST

314.5"   314.5"   62"

MAXIMUM ALLOWABLE
PARALLEL VERTICAL MOVEMENT OF #1 & #2 BEARINGS (ASE) = $\pm 12\frac{1}{2}$ MILS
NON-PARALLEL "   "   " " " "   "   " = $\pm 2\frac{1}{2}$ "
VERTICAL MOVEMENT OF #3 BEARING   = ± 10 "

1st GROUP OF SHIPS MODIFICATION

4    $22\frac{13}{15}$ BHD   2   1

THRUST

200.5"   368.5"   62"

MAXIMUM ALLOWABLE
PARALLEL VERTICAL MOVEMENT OF #1 & #2 BEARINGS (ASE) = ± 22 MILS
NON-PARALLEL "   "   " " " "   "   = ± 4 "
VERTICAL MOVEMENT OF #3 BEARING   = $\pm 14\frac{1}{2}$ "

2nd GROUP OF SHIPS MODIFICATION

4   3 $21\frac{7}{8}$ BHD   2   1

THRUST

260.5"   368.5"   62"

MAXIMUM ALLOWABLE
PARALLEL VERTICAL MOVEMENT OF #1 & #2 BEARINGS (ASE) = ± 24 MILS
NON-PARALLEL "   "   " " " "   "   = $\pm 4\frac{1}{2}$ "
VERTICAL MOVEMENT OF #3 BEARING   = ± 16 "

Figure 12

Span ratio (L/D) is used as a rough design guideline for shafting and is defined as the ratio of bearing center distance to shaft diameter. Reference [29] gives values of L/D varying from 12 minimum to 20-22 maximum. The original shaft design had a span ratio of 13 which increased to 16 after modification.

In later vessels, the use of higher strength material permitted a further reduction in shaft diameter and additional flexibility.

It is important to note that the flexibility of the original shaft design, although on the low side, fell within the guidelines based on past practice yet was not sufficient because of the increased flexibility of the machinery supports.

Relocating the main thrust bearing to a position aft of #3 bearing eliminated ninety per cent of the effect upon the gear of a bending moment in the shaft caused by tilt of the thrust bearing housing. In addition, the depression of the gear supporting structure upon application of thrust was eliminated by the increased distance from the gear and, perhaps more important, the bearing was positioned within the shaft alley which, with its sides, overhead deck, inner bottom and shell, formed a stiff girder.

5.13 Main Machinery Foundations

The original arrangement of the propulsion machinery foundations is shown in Fig. 13. The main thrust bearing, located just aft of the reduction gear, was subject to a force of approximately 380,000 pounds at full power. This force was transmitted to the shell via two longitudinal thrust girders which served to spread the load to the tank top and the grid of longitudinal and transverse structure within the inner bottom over a fore and aft span of about 28-30 ft. The moment formed by the force and the distance to the basic hull was responsible for the deflection of the inner bottom, the consequent change in slope or rotary movement of the thrust foundation, and the depression of the gear foundations.

Longitudinal stiffness is required to resist the bending moment and is obtained most effectively by deep girders. Fig. 13 shows that it was necessary to reduce the depth of the thrust girders in order to pass forward under the bull gear; however, aft of the thrust bearing there were no obstructions and the extent of the girders was limited only by the designer's decision.

INITIAL FOUNDATION DESIGN

SHAFT ALLEY

MAIN
THRUST
BEARING
380,000 lbs EST.

CL SHAFT
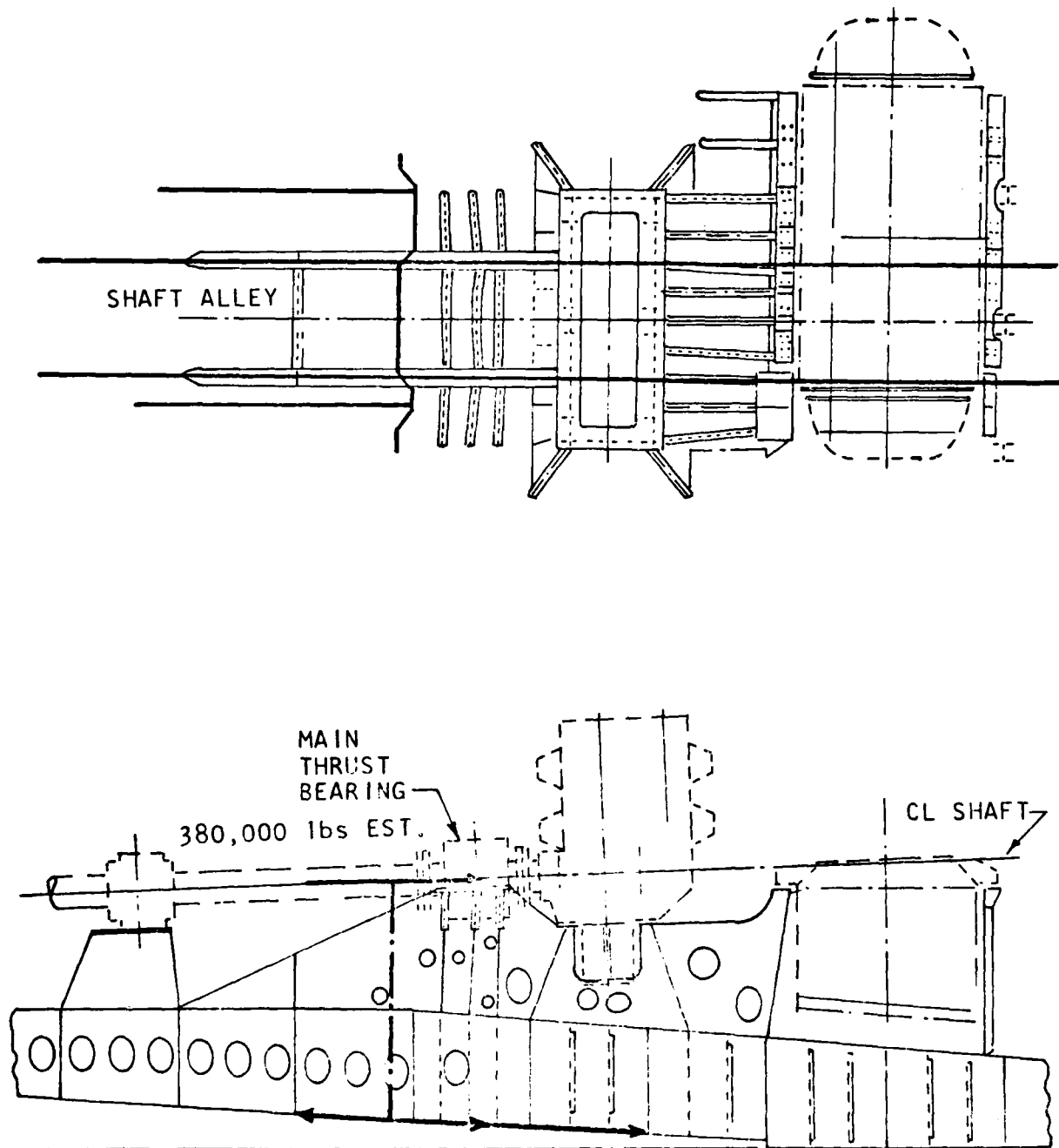
Figure 13

5.14 Structural Modifications

Changes were made to the eleventh vessel with the intent of generally stiffening the machinery supports and in particular reducing the tilt and twist of the gear foundations (Figs. 14 and 15). The extent of the changes was restricted by the practical difficulties of modifying an existing structural arrangement. Consequently, the limited changes which were made probably did not significantly reduce the deflections of the machinery supports. Additional structure was installed in the second group of ships while under construction including two complete floors under the gear foundation in an effort to provide increased resistance to deflection by torque forces (Fig. 16).

No measurements have been made on the vessels having additional structure and, therefore, it is not known to what degree these changes were effective.

5.15 Summary

Incompatibility between the flexibility of the hull structure and the rigidity requirements for support of the machinery caused failures in the reduction gears.

The flexibility of the main shaft, although at the low end of the allowable range, met existing design guidelines but was not sufficient to account for the hull structural guidelines.

No quantitative information was available regarding the flexibility of the hull structure during the design period and no measurements were made following structural changes. It does not appear that the structural changes alone would have been sufficient to correct the problem but instead the major portion of the improvement was due to the more flexible shafting and to the relocation of the main thrust bearing.

MODIFICATIONS TO STRUCTURE
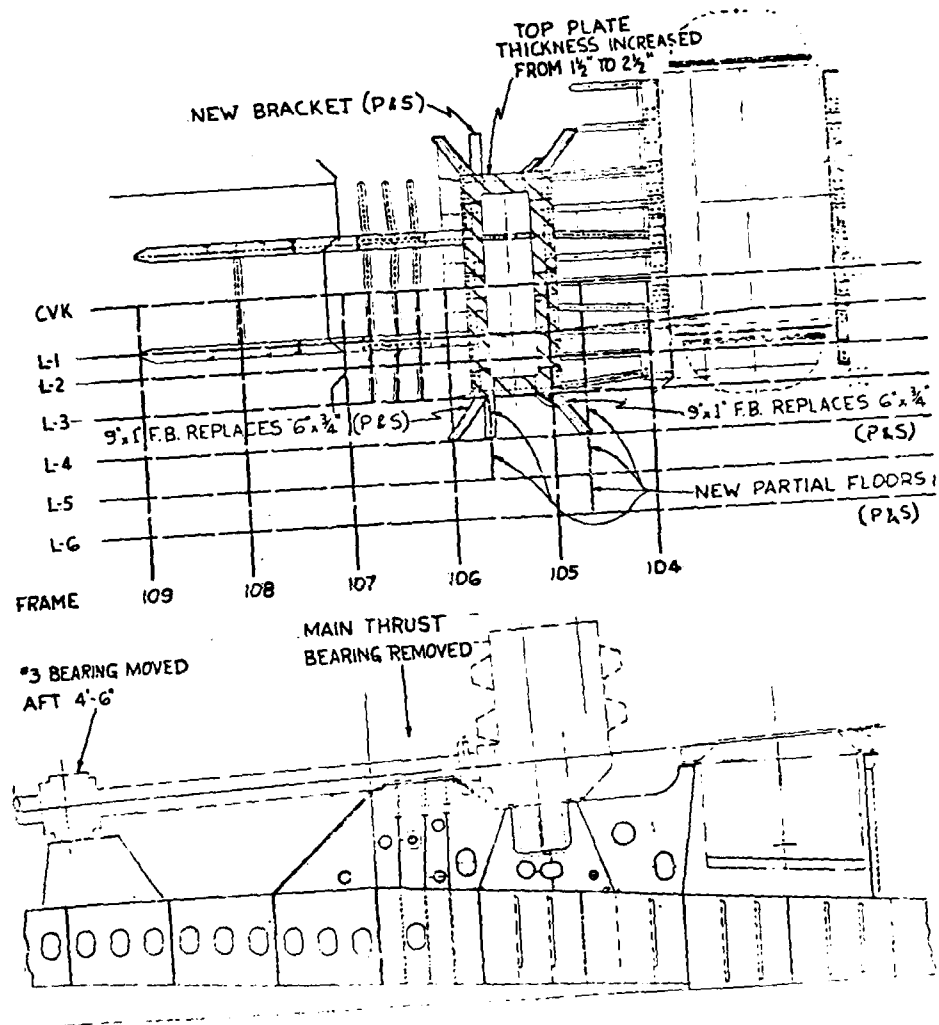
Figure 14

MODIFICATIONS TO STRUCTURE

REDUCTION GEAR FOUNDATION

FORWARD

FLANGE
INCREASED FROM
6"x3/4"TO9x1"

TOP PLATE THICKNESS
INCREASED FROM
$1\frac{1}{2}$"TO$2\frac{1}{2}$"

NEW   NEW

ORIGINAL PARTIAL FLOOR

AFT

9"x1"

NEW   NEW
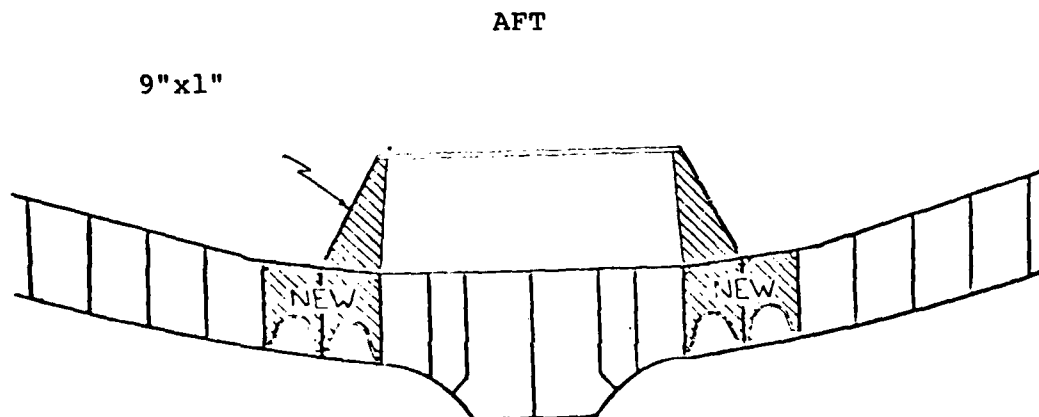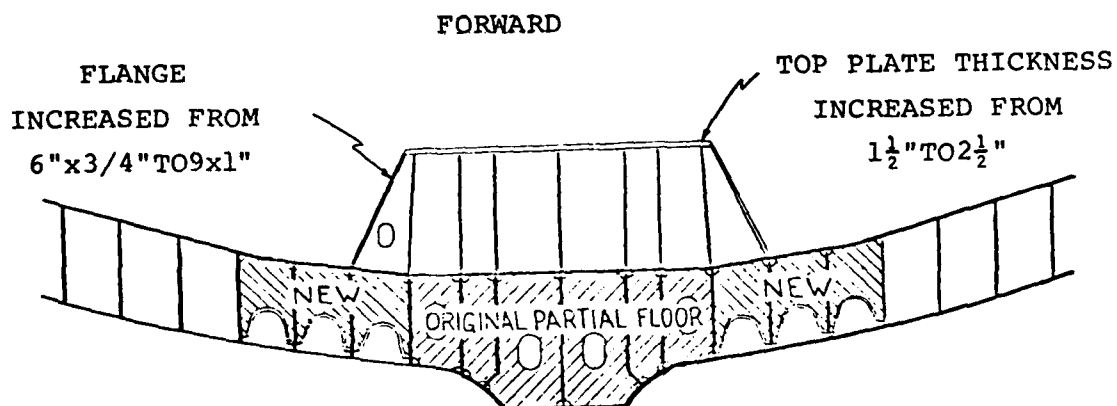
Figure 15

MODIFICATIONS TO STRUCTURE

2nd GROUP OF SHIPS

MACHINERY SPACE INNER BOTTOM

TWO ADDITIONAL FLOORS SUPPORT THE GEAR FOUNDATION
AND RESIST DEFLECTION DUE TO TORQUE
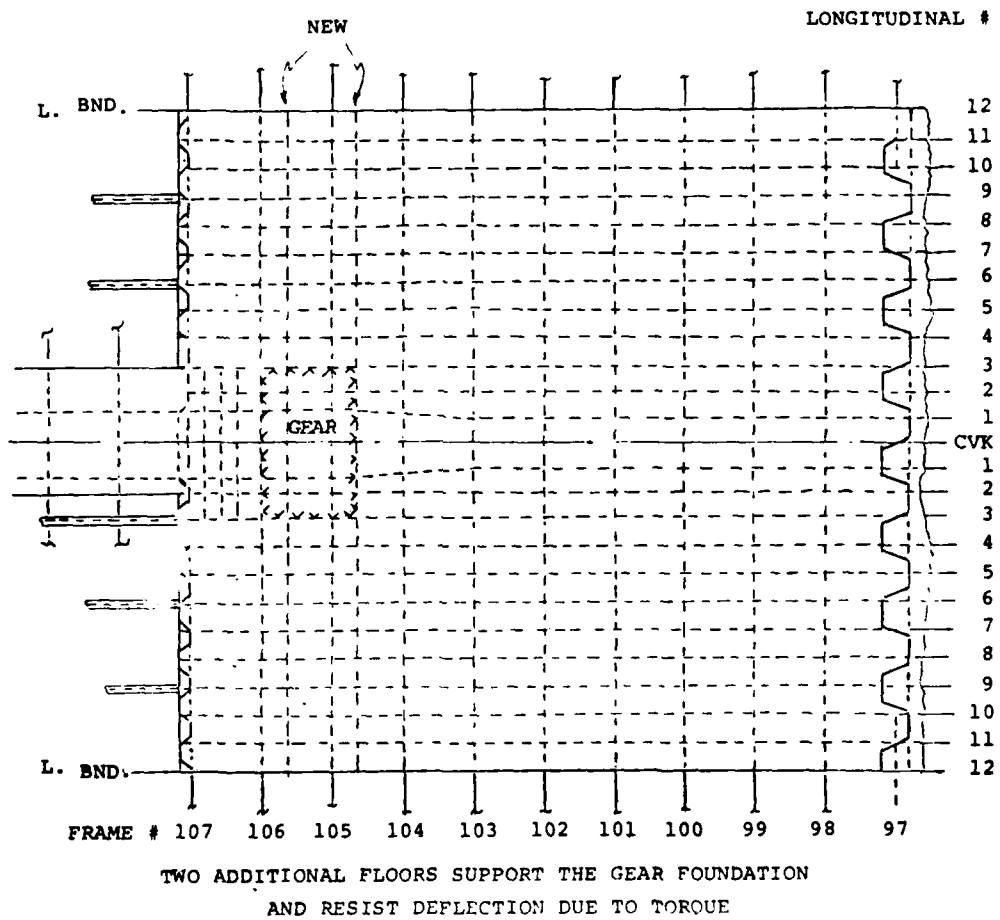
Figure 16

CHAPTER II. FOUNDATION DESIGN

1.   Review of Machinery Spaces Structural Arrangement

1.1 Relevant Structural Parameters

In order to study the structural design of machinery spaces of large ships, it is convenient to discuss first the main parameters which characterize the design. Essentially, four groups can be defined, as shown in Table II (refer to Fig. 17). The first four describe the types of structural members present: transverse, longitudinal, vertical and plane. The transverse, longitudinal and vertical members are essentially prismatic, while the plane members are essentially plated structures. The fifth group includes what we may call load-related parameters.

Structural details are not considered here for various reasons. Firstly, because they do not contribute to a large extent to the overall stiffness of the machinery spaces, which is the point of main concern here. Secondly, because it is assumed that structural details are properly designed in order to insure proper joining of the various components, good structural continuity, and in order to avoid stress concentrations and local instabilities such as tripping. Finally, the whole area of ship structural details has already been the subject of extensive research sponsored by the Ship Structure Committee [31,32], so that there is no need to consider it here.

Transverse members include frames, floors and web frames. The important parameters which define frames are spacing and scantlings (web thickness and depth, flange thickness and depth). The side shell can be assumed to be attached to each frame providing an effective breadth based on any acceptable theoretical approach, such as the ones reviewed in [33]. Web frames can have a rather complex geometry, particularly towards the ship ends, and as such cannot easily be defined by a small set of parameters.

Floors can essentially be defined by the average thickness t_f and the location, or the number i_f of frame spacings separating them, assuming a uniform spacing is used throughout the machinery spaces.

Longitudinal members essentially include the bottom center girder, bottom side girders and stringers. The center girder can be defined by the depth d and thickness t_C. The side girders have in general the same depth as the center girder, so that the main parameters are the number, location and thickness of t_s. In addition, bottom girders can be stiffened in order to prevent sideway or instabilities, and this obviously makes the structure's description more complex.

TABLE II

MAIN STRUCTURAL PARAMETERS

1. Transverse members

    a.   frames (s, I)

    b.   floors (i_f, t_f)

    c.   web frames (i_w, I_w)

2. Longitudinal members

    a.   center girder (d, t_{cg})

    b.   side girders (i_{sg}, t_{sg})

    c.   stringers (s_s, I_s)

3. Vertical members

    a.   stanchions

4. Plane Members

    a.   inner bottom (t_i)

    b.   intermediate decks

    c.   longitudinal bulkheads

    d.   transverse bulkheads

5. Load-related parameters

    a.   point of application of large weights

    b.   thrust bearing above base (H)

MAIN STRUCTURAL COMPONENTS
OF MACHINERY SPACE
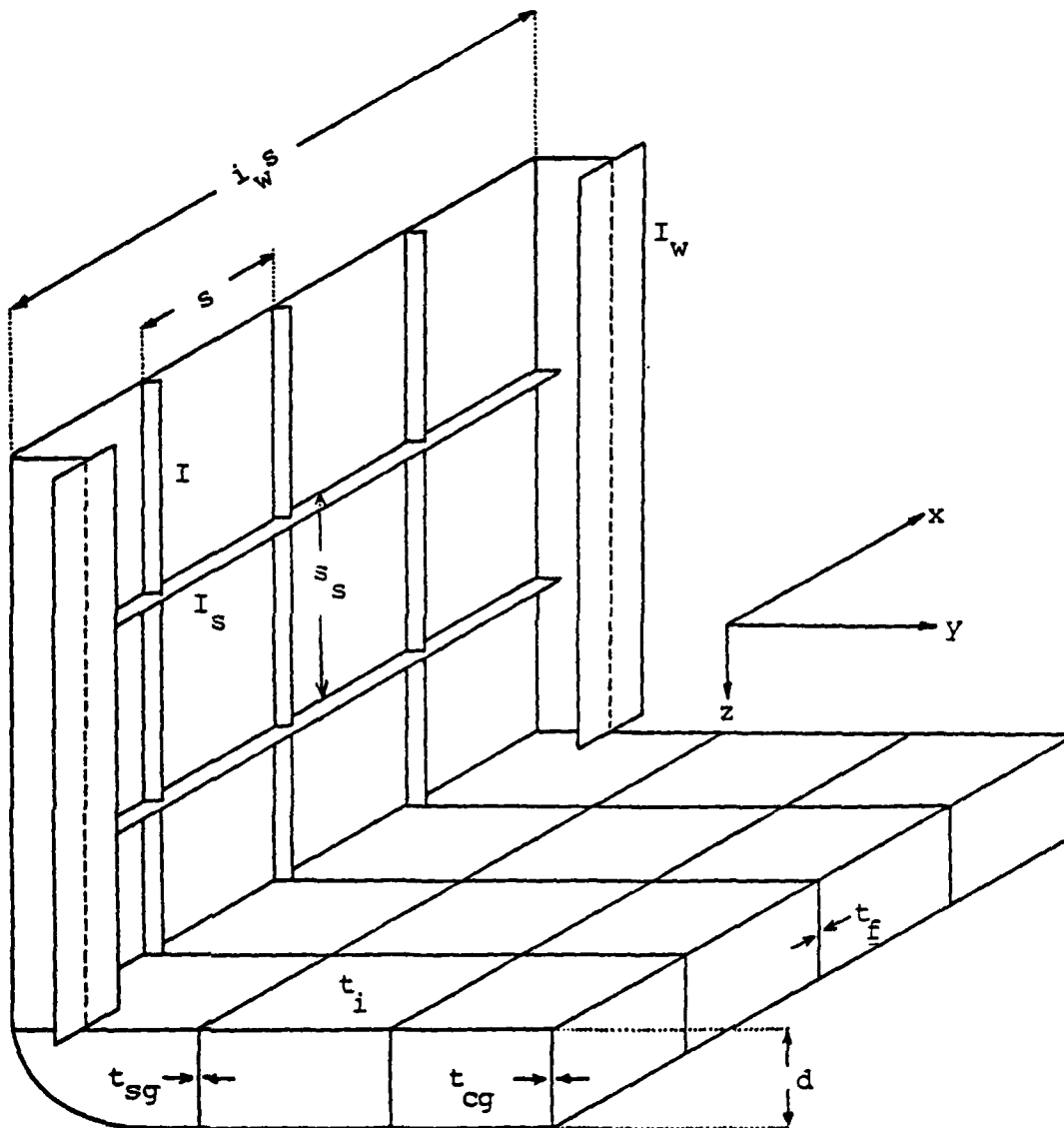
Figure 17

In order to define the stringers, the location and scant-
lings have to be given. The scantlings include web depth and
thickness and flange width and thickness.

Vertical prismatic members are essentially stanchions for
which scantlings and location have to be defined.

Plane members include the inner bottom defined by the
thickness and stiffening arrangements, intermediate decks or
flats and bulkheads. The intermediate decks in the machinery
spaces are normally made of orthogonally stiffened panels and
they have, in general, large openings. Bulkheads are also,
in general, made of orthogonally stiffened panels. Thus, the
geometry of intermediate decks and longitudinal bulkheads is
not easy to define by a small set of parameters.

The load-related factors in the case of the machinery space
include the points of application of large weights, such as
the weights of machinery components (turbines, boilers, con-
densers, reduction gears, etc.) and tanks. These are fixed
for a given design and the designer cannot in general modify
them. A very important load-related parameter is the height
H of the thrust bearing above its foundation base. It is
obvious that this height has a minimum permissible value. In
some designs, the tank top is penetrated by the gear but there
is a limit on how deep this interference can be. As mentioned
in Chapter I, the thrust-bearing height above the inner bottom
essentially provides a cantilever effect to the applied thrust,
which is a very important load acting upon the foundation. As
the height increases, the moment transmitted to the foundation
becomes larger, and this is one of the major causes for the
tilting of the reduction gear and associated failures. Thus,
when studying the machinery foundation stiffness, this is a
parameter which must certainly be considered.

We can conclude that in order to describe the structural
arrangement of the machinery space a very large number of geo-
metrical and structural parameters have to be defined.

The transverse frames are defined in terms of spacing s
and moment of inertia I about the x axis (see Fig. 17),
assuming they only provide a significant stiffness in the yz
plane, and that they are equal and equally spaced. Instead
of the moment of inertia, the section modulus could obviously
be used. However, since our main concern here is the stiffness
rather than the strength, the moment of inertia is the most
adequate parameter. If the moment of inertia is fixed, the
scantlings can be determined from simple design rules governing
proportions, such as web depth/web thickness, as suggested, for
example, in [10]. Similarly, the web frames are defined in
terms of frame spacings i_w separating them and moment of inertia
I_w, and it is again assumed that they are equal and equally
spaced. Since the web frame geometry can vary largely in the

vertical and longitudinal directions, the moment of inertia can be specified at its lowest span between the inner bottom and the first flat, and essentially at midlength of the engine room space or closer to the reduction gear casing.

As stated earlier, the floors are defined in terms of the number of frame spacings i_f separating them and thickness t_f, and it is also assumed they are uniform and evenly spaced.

The center girder is determined by the depth d and average thickness t_c. The side girders are assumed to be equal and equally spaced between the side and the center girder, so that they can be defined by their number i_{sg} and average thickness t_{sg}. The stringers can also be assumed to be equal and equally spaced by an amount s_s and they are characterized by a certain moment of inertia I_s about the y-axis, since they only provide a significant stiffness in the xz plane (see Fig. 17).

The stanchions can be assumed to be rigid, since their axial stiffness is large and in practice they are designed to preclude the possibility of buckling. In machinery spaces, stanchions are usually used to provide support to large local weights, such as the boiler, or to provide support to deckhouse or superstructure ends. As such they can serve as vehicles to transmit to the foundation large concentrated loads which can induce important deflections. Thus, they should not be neglected when carrying the structural analysis of the machinery spaces. For convenience, they can be taken as rigid struts acting at well-defined locations, and we can assume here that the designer has no freedom in changing their number.

The inner bottom can be defined by an average thickness t_i. In reality, the inner bottom is also a fairly complex structure if all the structural details and stiffening members are taken into account. The problem is simplified here by defining it only in terms of the thickness t_i, and assuming that the stiffening members such as beams and longitudinals can be associated to frames and bottom girders, respectively.

Intermediate decks and bulkheads cannot be treated in detail in any simple mode. The intermediate decks essentially provide lateral support to the side shell, so that the important factors are the number and location in the vertical direction, say height above the inner bottom. Similarly, the longitudinal bulkheads provide support to the bottom shell and decks and as such can be characterized in terms of number and location in the horizontal direction, say distance to the centerline. Transverse bulkheads need not be considered here since, in general, they are only used at the forward and after ends of the machinery space. It should be noted that, in general,

the designer does not have very much freedom in choosing the
number and locations of bulkheads or intermediate decks, since
these are fixed by other considerations such as general
arrangement, subdivision requirements, or machinery space
requirements. Thus, it can be assumed, when studying the machin-
ery space structural arrangement, that these are given. The
designer should then make sure that they are properly stiffened
so that they can provide the necessary support to the structure.

If this simplified model of the structural arrangement of
the machinery space is adopted, then the foundation stiffness
can be expressed as a function of a well-defined number of
parameters, as follows:

Stiffness = function (frames, floors, web frames, center
girder, side girder, stringers, inner bottom,
thrust bearing above inner bottom)

$$= \text{function } (s,\ I,\ i_f,\ t_f,\ i_w,\ I_w,\ d,\ t_{cg},\ i_{sg},\ t_{sg}$$
$$s_s,\ I_s,\ t_i,\ H)$$

Thus, the stiffness is essentially a function of the four-
teen parameters listed above. While the first thirteen can be
to a large extent controlled by the designer, the last one H
is usually determined from considerations other than stiffness
or strength, since it essentially depends on the gear general
dimensions and geometry.

In Chapter IV, the model just described is used to perform
some parametric variations on the structural design of the
machinery space of a given ship, in order to extract some use-
ful information on the best way of meeting the recommended
stiffness requirements.

1.2 Review Summary

In order to obtain a reasonable description of current
design practice concerning the structural arrangement of mach-
inery spaces, the drawings of various ships were studied and
compared, and this section summarizes the most relevant findings
of this task. Some of the conclusions reached were used to
prepare the set of design recommendations given at the end of
this Chapter.

The main particulars of the ships studied are given in
Table III. The table includes ship type, deadweight, main
dimensions (length between perpendiculars, beam, depth and
draft), machinery type and installed horsepower. Each ship
is identified by the number given in the left hand column of
Table III. Ships #1 through #13 are geared steam-turbine
powered, and #14 through #21 are diesel powered. Within

TABLE III

SHIP MAIN CHARACTERISTICS

#	Type	Δ (1)	Main Dimensions (2)				Mech. Type	HP
			L	B	D	T		
1	LNG	125,000	897	144	82	36	Steam	43,000
2	LNG	125,000	906	135	85	36	Steam	40,560
3	LNG	125,000	887	141	94	38	Steam	40,000
4	Tanker	269,574	1050	179	89	69	Steam	40,000
5	Tanker	276,424	1063	178	88	69	Steam	38,000
6	Tanker	290,800	1095	188	94	67	Steam	36,000
7	Tanker	265,000	1060	178	86	67	Steam	35,000
8	Tanker	264,197	1050	176	87	67	Steam	34,000
9	Tanker	249,550	1080	170	84	66	Steam	32,000
10	LASH	40,311	797	100	60	38	Steam	32.000
11	Tanker	188,500	915	166	78	59	Steam	28,000
12	Tanker	164,000	864	173	79	57	Steam	26,700
13	Tanker	120,000	850	138	68	52	Steam	26,000
14	Container	29,194	679	106	62	34	Diesel	36,000
15	Ro-Ro	25,000	652	106	67	33	Diesel	26,800
16	Tanker	135,000	833	143	75	56	Diesel	26,100
17	Tanker	81,283	735	133	43	65	Diesel	15,120
18	Tanker	52,068	658	105	58	39	Diesel	15,000
19	Bulk C.	60,000	702	106	58	40	Diesel	14,000
20	Bulk C.	64,657	715	106	43	60	Diesel	12,960
21	Bulk C.	34,400	548	91	49	37	Diesel	12,600

(1) Δ denotes the deadweight for all ships except LNG's where it denotes the capacity in cubic meter.

(2) L,B,D and T are respectively the length between perpendiculars, breadth, depth and draft in ft (rounded).

each one of these two groups, the ships are listed in decreasing
order of installed horsepower. Within the diesel-powered ships
group, only #18 and #19 have independent reduction-gear units.

The only two ships that are specifically considered in
this study are the LASH #10 and the tanker #11. The LASH
vessel was already discussed at length in Chapter I, while the
tanker is the subject of detailed calculations described in
Chapter IV, which serves as an illustration of the method being
proposed here. There are several reasons for concentrating
our efforts in these two particular vessels. These two ships
have considerably different afterbody shapes, the tanker having
a full or "U" form and the LASH an open form. Furthermore,
the machinery space configuration for the tanker is essentially
as shown in Fig. 1a, while for the LASH it is as shown in Fig. 1b.
For the LASH, considerable data were available and an interesting
case study could be presented. On the other hand, the tanker
has not developed any hull-machinery incompatibility problems,
and its conventional design can be considered as quite represent-
ative of current practice.

For each ship listed in Table III, shafting particulars
are given in Table IV. The identifying number in column 1 of
Table IV is the same as used in Table III. The main shaft
particulars included in Table IV are the total length of shafting
(length from reduction gear coupling to after end), the total
number of journal bearings, the mean diameter and length of each
shaft section, the distance from the point of support for all
bearings at centerline from the reduction gear after bearing
or diesel enginer coupling. In addition, the span/diameter
ratio, (L/D) for the shaft segment closer to the reduction
gear (or diesel engine) is also given. As noted in Chapter I,
this is an important parameter since it gives an indication
about the shaft stiffness close to the reduction gear, and as
such can have a strong impact on the magnitude of the reactions
at the gear bearings. In the geared vessels examined (#1
through #13, #18 and #19), the L/D ratio lies between 10.21
and 16.84, which is on the low side of the interval 12 minimum
to 20-22 maximum given in [29]. In the case of direct-drive
diesel engines, the ratio is substantially smaller, ranging
from 2.57 to 8.18.

Table V contains some main-machinery space characteristics,
including the engine room main dimensions (length, width aft
and width forward), the double-bottom depth and the thrust-
bearing height above the inner bottom. The engine room dimen-
sions refer to the tank top level. It can be concluded that
all the ships for which dimensions are given have an engine
room of trapezoidal shape, such as sketched in Fig. 1a, except
#10 for which the engine room essentially has a uniform width
and a square shape as in Fig. 1b.

TABLE IV

SHAFT CHARACTERISTICS

#	L_T (ft) (1)	N (2)	Shafting Details (3)			Bearing Dtls (4)		L/D (5)
			Code	Mean Diam. (in)	Length (ft)	Code	D_1 (ft)	
1	63.50	2	IS	27.00	16.31	SF	37.13	16.50
			LS	27.00	22.00	SA	54.75	
			TS	33.00	16.19			
2	140.00	4	N/A	N/A	N/A	N/A		N/A
3	106.06	4	LSF	25.00	28.61	L	24.78	11.89
			LSA	24.75	31.31	L	45.44	
			T	32.25	38.25	L	71.44	
						SA	95.68	
4	67.79	3	IS	27.96	30.06	L	30.71	13.18
			TS	37.00	27.89	SF	45.88	
						SA	57.38	
5	61.68	3	IS	26.38	27.26	L	25.10	11.42
			TS	34.00	26.25	SF	41.67	
						SA	50.30	
6	68.89	3	IS	27.17	34.47	L	23.68	10.46
			TS	33.46	29.42	SA	44.31	
						SF	57.30	
7	64.79	2	IS	26.01	28.94	L	33.29	15.36
			TS	32.50	29.85	SA	57.08	

(1) L_T = total length of shafting. (2) N = number of journal bearings.
(3) Code for shafting details: L = lineshaft, T = tail shaft,
I = intermediate shaft, S = solid, H = hollow, ST = stern tube shaft,
F = forward, A = aft.
(4) Code for bearing details: L = line shaft br., SF = stern tube
forward br., SA = stern tube aft br., STB = strut br., D_1 = distance
to reduction gear after bearing (or diesel engine coupling).
(5) L = distance between reduction gear after bearing (or diesel
engine coupling) and closest bearing, and D = corresponding shaft
diameter.

TABLE IV (Continued)

SHAFT CHARACTERISTICS

#	L_T (ft) (1)	N (2)	Shafting Details (3)			Bearing Dtls (4)		L/D (5)
			Code	Mean Diam. (in)	Length (ft)	Code	D_1 (ft)	
8	62.28	3	IS	25.71	26.02	L	25.00	11.67
			TS	32.48	28.94	SF	39.75	
						SA	52.75	
9	64.25	3	IS	25.79	27.36	L	21.94	10.21
			TS	32.88	27.89	SF	44.94	
						SA	56.21	
10	160.85	5	LSF	21.88	33.13	L	30.71	16.84
			LSA	23.57	31.48	L	49.91	
			ST	28.56	38.24	L	76.13	
			I	29.50	33.65	SA	103.69	
			T	29.50	18.14	STF	149.41	
11	64.20	3	LS	23.75	27.39	L	29.46	14.89
			TS	29.75	28.48	SF	36.96	
						SA	49.96	
12	62.80	2	IS	24.25	26.72	L	28.19	13.95
			TS	32.50	27.08	SA	52.58	
13	72.88	3	IS	23.63	17.85	L	22.69	11.52
			LS	23.63	17.85	L	40.03	
			TS	30.69	28.85	SA	61.42	
14	120.31	6	IS	25.00	37.73	L	7.22	3.47
			LS	25.00	39.93	L	26.58	
			TS	33.00	42.65	L	45.94	
						L	65.30	
						SF	84.66	
						SA	109.59	

(1) - (5) See footnotes in previous page.

TABLE IV (Continued)

SHAFT CHARACTERISTICS

#	L_T (ft) (1)	N (2)	Shafting Details (3)			Bearing Dtls (4)		L/D (5)
			Code	Mean Diam. (in)	Length (ft)	Code	D_1 (ft)	
15	79.48	4	IS	20.28	9.20	L	13.83	8.18
			LS	20.28	35.00	L	33.02	
			TS	27.01	35.27	SF	49.51	
						SA	69.87	
16	43.11	3	IS	22.44	20.47	L	10.25	5.48
			TS	26.77	22.64	SF	25.72	
						SA	34.09	
17	46.18	4	IS	18.70	25.51	L	5.00	3.21
			TS	24.80	20.67	L	20.75	
						SF	30.79	
						SA	38.24	
18	35.42	2	IH	20.00	16.69	SF	22.53	13.52
			TH	26.00	18.73	SA	29.89	
19	38.16	3	IS	19.06	18.80	L	19.31	12.16
			TS	23.54	19.36	SF	25.91	
						SA	34.52	
20	40.08	4	IS	17.72	20.67	L	3.79	2.57
			TS	22.00	19.41	L	16.91	
						SA	25.60	
						SA	32.64	
21	39.73	3	IS	17.72	19.72	L	10.14	6.87
			TS	22.09	20.01	SF	23.71	
						SA	32.81	

(1) - (5) See footnote in previous page.

TABLE V

MACHINERY SPACE CHARACTERISTICS

#	Engine Room Dimensions (ft)			Double Bottom Depth (ft)	Thrust above inner bottom (ft)
	length	width aft	width fwd		
1	100.0	N/A	N/A	13.33	4.33
2	97.5	N/A	N/A	9.44	N/A
3	100.0	16.0	106.0	14.25	4.00
4	121.0	7.2	84.6	14.76	5.97
5	106.3	9.8	82.7	12.52	7.09
6	115.9	8.2	88.6	13.12	9.43
7	105.0	8.0	86.0	13.42aft 9.00fwd	3.50
8	106.3	8.5	69.9	13.34	8.75
9	93.8	23.0	69.2	8.14	9.84
10	73.3	70.5	70.5	8.75	7.00
11	107.5	8.3	90.0	9.00	9.50
12	95.0	11.0	68.0	11.00	6.50
13	93.0	9.7	72.7	13.00aft 8.00fwd	5.33
14	100.4	24.9	85.3	9.18	5.74
15	120.2	6.9	50.0	6.53	5.11
16	102.2	8.9	82.7	9.15	4.92
17	110.2	7.9	65.6	8.55	4.85
18	76.1	8.9	56.7	6.56	5.90
19	71.8	7.5	59.0	6.86	6.07
20	86.6	9.2	63.0	8.05	4.85
21	81.3	8.0	68.9	7.12	5.00

The double-bottom depth for the steam-powered ships varies from 8.14 to 14.76 ft. For the diesel-powered ships, the depth is, in general smaller, varying from 6.50 ft. to 9.18 ft.

The thrust-bearing height above the inner bottom varies from 3.50 ft. to 9.84 ft. for the steam-powered ships. For the direct-drive diesel-engine ships, it varies from 4.85 ft. to 5.75 ft. For the geared diesel-powered ships #18 and #19, it is equal to 5.90 ft. and 6.07 ft. respectively. In those cases where the height is relatively large, the cantilever effect discussed earlier is reduced by having a sloping tank top which ensures a very gradual transition between the inner bottom and the thrust bearings.

The most relevant information regarding the structural arrangement of the machinery spaces of the ships listed in Table III are summarized in Tables VI and VII.

The frame spacing in way of the machinery spaces can be for convenience expressed in terms of the length between perpendiculars L, by computing the ratio between the frame spacing in inches and L in feet and multiplying by 100.

This ratio is given in the third column of Table IV. For vessels #1 through #8, #11 and #12, this ratio lies between 3.15 and 3.47. For #9, it is equal to 2.92; for #10, it is equal to 11.04 and for #13 to 4.24. For all the diesel-powered ships, the ratio lies between 4.02 and 5.75. Thus, we can conclude that for most steam-powered ships the ratio is smaller than 3.50. For one ship out of thirteen, the ratio is slightly larger and equal to 4.24, while for #10 it is much larger and equal to 11.04. This larger value is due to the fact that the LASH vessel has a strong web at every frame in way of the machinery space. It also has closely spaced stringers or side longitudinals implying a predominantly longitudinally framed structure. All diesel-powered ships have a larger ratio than steam-powered ships (excepting #10, and #16 for which the ratio is 4.02, smaller than what it is for #13).

Regarding floors, all ships have one floor at every frame, except #9 for which the floor spacing in terms of frame spacing can vary between 1 and 3 in the machinery space.

The web spacing in terms of frame spacings varies for most ships between 3 and 5. Ship #10 has web frames at each frame.

The stringer spacing for nine ships is smaller than 40 inches, indicating a clear longitudinal framing arrangement. For eight ships, it lies between 60 and 150 inches and for three ships, it is larger than 200 inches indicating, in this case, a predominantly transverse framing.

The number of side girders is given in way of the reduction gear or engine coupling in the last column of Table VI. For all ships, it varies from 4 to 10, except for #10 where it is equal to 20, which is due to the large span this vessel has at the tank top in way of the machinery spaces.

Table VII lists the average thickness of center girder, side girders, floors and inner bottom in way of the reduction gear or engine coupling. The center girder thickness varies for all ships between 0.57 and 1.00 inches. The thickness of side girders varies from 0.55 to 1.46 inches. This is, in fact, a parameter which is difficult to determine since it can vary quite widely, depending among other factors on the distance to the ship's centerline. The average floor thickness for all steam-powered ships varies from 0.63 to 0.79 inches, except for #10 for which it is equal to 0.56 inches and #13 for which it is 0.60 inches. The inner bottom average thickness varies from 0.57 to 1.10 inches for all ships.

Table VIII lists the moment of inertia in in.^{4} for frames, web frames and stringers. Referring to Fig. 17, the moment of inertia is given about an axis parallel to the x-axis for frames and web frames, and parallel to the y-axis for stringers. This assumes that frames and web frames essentially bend in the yz plane, while stringers essentially bend in the xz plane. In all cases, the scantlings were taken for members as close as possible to the reduction-gear casing, or engine coupling. As stated earlier, the geometry of web frames can be quite complex, and the figures given correspond to the frame scantlings closer to the point of interest mentioned above. For frames and web frames, shell plating with a width equal to the frame spacing was included in the computation of the moment of inertia given in Table VIII. For stringers, no shell plating is included in the moment of inertia. It can be concluded from Table VIII that the frame moment of inertia, for those ships for which it could be computed, lies between 175 and 2,597 in^4. The web frame inertia lies between 8,166 and 128,006 in^4. The stringers' moment of inertia lies between 68 and 20,934 in^4.

2. Recommendations for Machinery Foundation Design

2.1 General Guidelines

The term "foundation" is often limited to structure above the inner bottom which directly serves to support the machinery; however, to be fully effective the foundation must be properly integrated with the structure of the basic hull and become a part of an overall system.

TABLE VI

MACHINERY SPACE STRUCTURAL PARAMETERS

#	Frame spacing (in)	s/L (1)	Frame spacings between		Stringer spacing (in)	Number of side girders
			floors	web frames		
1	30.00	3.34	1	4	60.00	6
2	30.00	3.31	1	N/A	N/A	6
3	30.00	3.38	1	8	228.00	6
4	35.43	3.37	1	3 and 4	27.56	8
5	35.43	3.33	1	4	230.40	10
6	34.44	3.15	1	4	35.40	8
7	36.00	3.40	1	3 and 4	81.48	6
8	35.43	3.37	1	2 to 4	35.40	6
9	31.50	2.92	1 to 3	3	149.52	6
10	88.00	11.04	1	1	36.00	20
11	30.00	3.28	1	3 to 5	90.00	10
12	30.00	3.47	1	4 and 5	105.00	8
13	36.00	4.24	1	3 and 4	78.00	8
14	35.43	5.22	1	3	33.46	4
15	30.71	4.71	1	3	78.72	4
16	33.46	4.02	1	4	32.64	10
17	31.50	4.29	1	4	125.98	4
18	31.50	4.79	1	4	23.62	4
19	31.89	4.54	1	3	23.64	6
20	31.50	4.41	1	4	379.92	6
21	31.50	5.75	1	3	118.08	6

(1) s/L = frame spacing in inches divided by the
    length between perpendiculars in feet, multiplied
    by 100.

TABLE VII

MACHINERY SPACE STRUCTURAL PARAMETERS

#	Thickness (in)			
	center girder	side girders	floors	inner bottom
1	1.00	0.75	0.75	0.88
2	0.88	0.63	0.63	0.75
3	0.81	0.75	0.63	0.88
4	0.87	0.71	0.71	0.98
5	0.94	1.46	0.71	0.94
6	0.98	0.98	0.79	1.10
7	0.94	0.75	0.75	0.86
8	0.85	0.69	0.69	0.93
9	0.98	0.98	0.71	0.87
10	0.75	0.56	0.56	0.75
11	0.81	0.63	0.63	0.72
12	0.75	0.75	0.63	0.75
13	0.81	0.81	0.60	0.79
14	No C.G.	1.48	0.55	0.75
15	0.63	0.79	0.63	0.79
16	0.98	0.67	0.59	0.69
17	0.75	0.96	0.71	0.63
18	0.59	0.59	0.59	0.65
19	0.79	0.98	0.71	0.65
20	0.69	0.55	0.71	0.65
21	0.57	1.02	0.79	0.57

TABLE VIII

FRAME, WEB FRAME

AND STRINGER INERTIAS

#	Moment of inertia (in^4)		
	frames	web frames	stringers
1	907	8,166	
2			
3	175	14,698	5,440
4		128,006	
5		75,812	15,851
6			466
7	2,597	55,727	20,934
8		111,026	475
9	1,901	74,856	4,723
10			179
11	426	23,630	3,336
12	1,143	17,516	8,359
13	1,226	45,893	7,712
14		52,990	
15	311	14,692	1,952
16	No	35,399	415
17	930	25,536	2,462
18	No	30,180	366
19	No	17,179	68
20	1,098	17,369	2,308
21	494	11,731	1,578

Foundations must (1) transfer the propeller thrust and torque to the basic hull and (2) maintain proper alignment of the machinery components.

The main-thrust bearing (1) transfers propeller thrust to its foundations and (2) positions the main-shafting system longitudinally.

Propeller thrust consists of two components (1) steady force and (2) periodically varying force caused by the variable wake in which the propeller blades operate.

Generally, maximum resistance of the thrust bearing foundation to deflection by thrust forces, both static and dynamic, is desirable and results in minimum movement at the bull gear. In certain past cases where a longitudinal natural frequency occurred in the operating range near full power, attempts to lower the frequency by softening the thrust foundation caused the vibratory amplitude at the bull gear to increase beyond safe limits for satisfactory operation of the gear teeth and instead required adjustment of one or more of the other variables.

Symmetrical support of the thrust housing, by using its port and starboard horizontal flanges, sometimes called "center-line support", avoids deflection of the housing which may affect the equal loading of all thrust shoes despite the use of equal-izing links or spherical support rings. This also eliminates the bottom vertical support whose bolting to the foundation is generally difficult to access.

The thrust force and distance from the bearing to the basic hull form a large moment applied to the structure. Accordingly, shaft rake should be kept as small as possible, consistent with the configuration of the reduction gear and condenser, in order to minimize the moment arm.

Longitudinal girders are provided to carry the thrust force from the bearing to the basic hull. The primary girders are located as close as possible to the thrust bearing and ex-tended fore and aft to spread the force and moment over as much hull structure as possible. Maintain the girder depth to resist deflection due to the bending moment; however, depth must be reduced as the girders pass under the bull gear. To compen-sate for the loss of stiffness, secondary girders may be pro-vided to serve a dual purpose. When arranged to line up with the outboard side of the reduction-gear case, they support the gear case and there is no loss of depth. Whenever practical, thrust girders should be extended fore and aft between bulkheads, since each bulkhead provides a vertical anchor point.

Provision must be made to transfer thrust force to the secondary girders by as direct a route as possible. This can be accomplished as shown in Fig. 18, which illustrates a well-engineered foundation. This foundation system was installed in several container ships having 32,000 SHP geared turbine machinery in an aft machinery space. The main thrust bearing is located immediately aft of the reduction gear, an arrangement that usually allows better foundation design and is particularly appropriate for high-powered installations.

The configuration of the centerline and two primary thrust girders is shown in the elevation view of Fig. 18. These girders are carried aft about a dozen frames at essentially full depth; however, forward of the thrust bearing the depth is reduced by both the gear and condenser. The secondary girders are similar except for full depth in way of the gear. In addition to distributing thrust forces, the secondary girders must resist deflection by torque reaction forces from the gear case which are upward on one side and downward on the other.

A portion of the total thrust is transmitted to the secondary girders by (1) sloping plates shown in plan "BB" and section "CC" and (2) a 1-1/2 inch thick horizontal plate below the thrust bearing.

The horizontal plate extends (1) forward to include the reduction gear and the aft turbine and condenser foundations, and (2) aft to include the first lineshaft bearing. The plate, therefore, positions these components in the athwartship direction and minimizes the possibility of athwartship mis-alingment or relative vibration. Finally, the plate is extended to the shell both port and starboard, which in the case of this vessel are relatively near because of converging waterlines, and therefore, additional paths are available to carry thrust to the shell. Tying to the shell is not always possible and depends upon the location of the machinery space and the hull shape at the stern.

Fig. 19 shows a well-developed foundation for a 36,000 SHP unit in a different type of hull. Here, the stern form consists of a broad flat transom and a relatively flat bottom rising toward the stern with the propeller-shaft-bearing strut supported. The machinery space is almost square in plan and the tank top essentially a large flat panel. With this config-uration, excitation of the inner bottom by the alternating component of the thrust force should be avoided by placing the main thrust bearing in way of, or near, a transverse bulkhead.

WELL-ENGINEERED FOUNDATION

LONGITUDINAL

SECTION "CC"

PLAN "BB"

PLAN "AA"

TANK TO

1 " PLATE EXTENDS TO SHELL PORT & STBD.

MAIN REDUCTION GEAR

MAIN THRUST BEARING

MAIN CONDENSER

B.L.

Figure 18

55
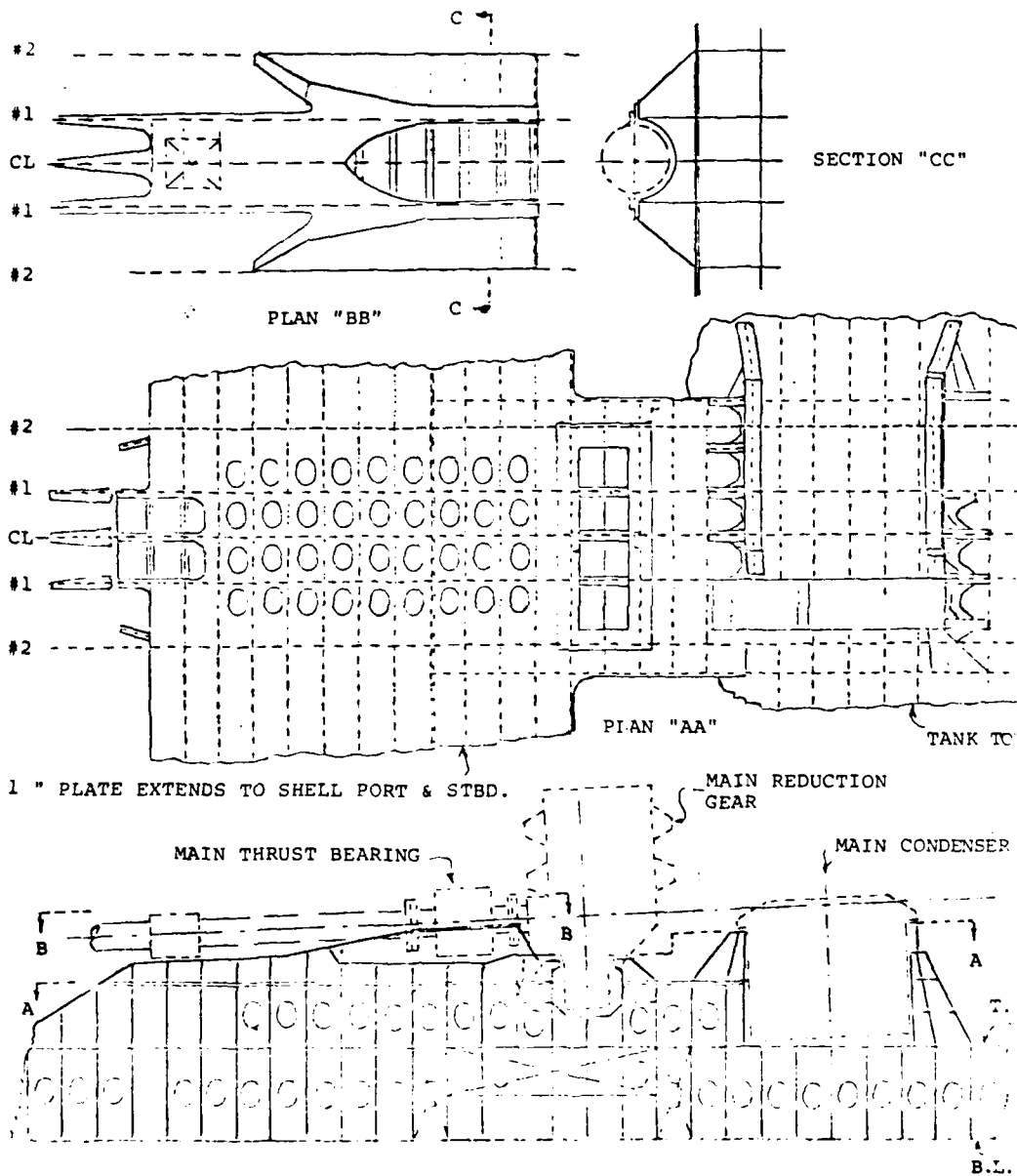
MAIN THRUST-BEARING FOUNDATION
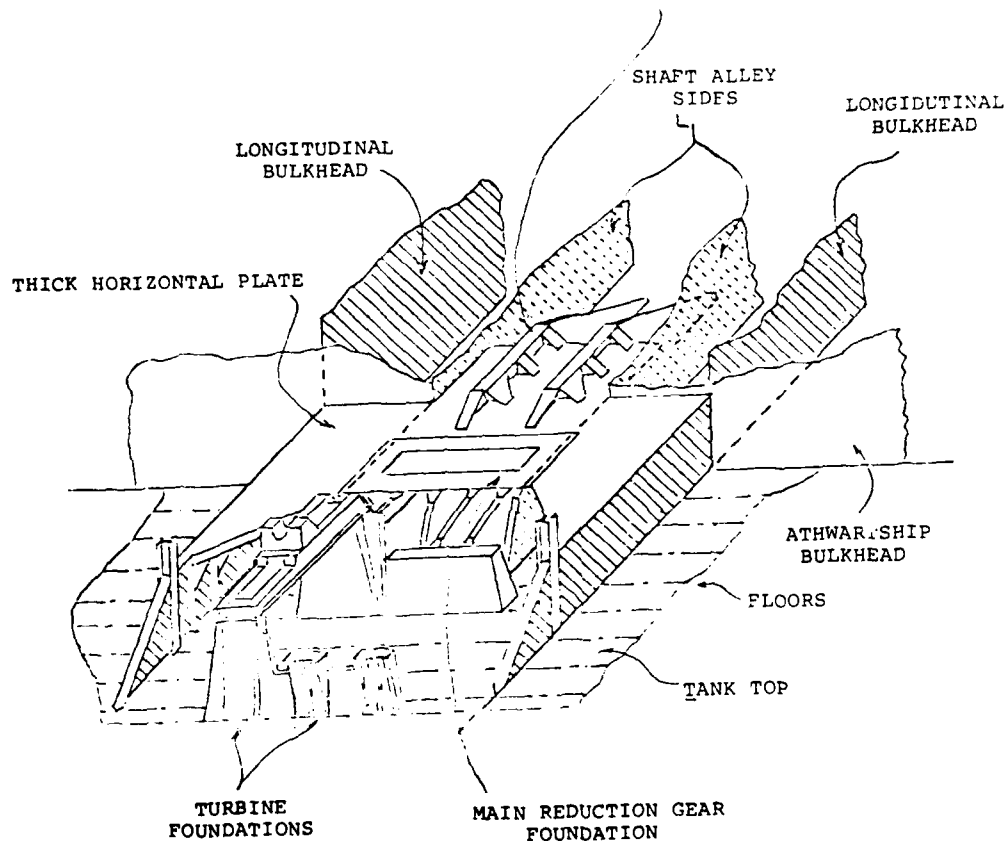
Figure 19

Fig. 19 illustrates excellent use of the ship's basic structure. The thrust bearing is located in a recess of the machinery space at the aft bulkhead. Seven thrust girders are used in conjunction with a thick horizontal plate. The shaft alley sides have been extended forward to carry the outboard sides of the gear. In addition, the horizontal plate is tied at its outboard edges to extensions of the deep longitudinal bulkheads.

Utilization of the stiffness of the shaft alley has been carried out successfully in a number of cases, either by placing the thrust bearing in the shaft alley, or by extending the sides forward as necessary.

Turbines are generally supported at their aft ends by extensions of the gear foundation. The forward turbine supports must maintain the position of the turbine in the vertical and athwartships directions and also provide flexibility in the fore and aft direction to allow for expansion of the turbine casings at increased operating temperatures. To maintain athwartship position, the forward turbine foundation should extend at least between the two secondary-thrust girders, or their extensions in the inner bottom. A separate support structure for the high-pressure turbine having a large height/athwartship width ratio should be avoided. Such a tall, narrow structure is sensitive to changes in the slope of the hull at its base and the movement is magnified by the height so that significant displacement of the turbine may occur with resulting misalignment at the turbine/gear flexible coupling. Several cases have been recorded where coupling or first-reduction gear problems have been due to deflections of the basic hull in an athwartship plane caused by changes in the drafts of the vessels.

Finally, consideration must be given to longitudinal bending of the basic hull and its effect upon alignment of turbine/gear, gear/line shaft, and line shaft bearings.

2.2 Classification Society Rules

An important source of design recommendations concerning the structural arrangement of machinery spaces is provided by the Rules of the Classification Societies, since they are, in general, based on extensive past experience. For this reason, various Rules have been reviewed, and the most relevant aspects of this review are summarized in Appendix A.

The following Classification Societies are included in this study:

    (a)   American Bureau of Shipping [34]
    (b)   Lloyd's Register of Shipping [35]
    (c)   Det norske Veritas [36]
    (d)   Bureau Veritas [37]
    (e)   Germanischer Lloyd [38]

The design recommendations provided by the Rules constitute part of the source material on which the summary that follows is based.

2.3 Summary

The following is a brief summary of the most important points regarding the structural arrangement of machinery spaces. They are based on the general guidelines of Section 2.1, on the review of ship drawings summarized in 1.2, on the recommendations included in the rules of the Classification Societies [34-40] and the literature [41-46].

a. It is very important that structural continuity be ensured. Particular attention should be given to the continuity of the double bottom in its connections to the structure forward and aft of the machinery space, and in way of the recess necessary for the installation of the main gear wheel.

b. The double bottom should provide enough stiffness (so that the proposed deflection limits for foundations of geared turbine propulsion units summarized on page 104 are satisfied) and be as deep as possible. If it is deeper than in other compartments, the transition should take place gradually beyond the aft and forward bulkheads of the machinery space, and any abrupt discontinuity should be avoided.

c. The thrust bearing foundations should act as much as possible in shear and tension, rather than bending and compression. The shaft rake should be kept small in order to reduce a "cantilever" type of effect.

d. Thrust girders should be extended fore and aft, if possible a few frame spacings beyond the transverse bulkheads which limit the machinery spaces.

e. Advantage should be taken of machinery casings and shaft tunnels in order to increase the stiffness of the structure in way of the machinery space.

f.   A thick horizontal plate below the thrust bearing should extend as much as possible in the fore and aft and athwartship direction to distribute thrust forces over a wide portion of the hull structure.

g.   Solid floor plates should be fitted along the line shafting and in the machinery space at every frame, and access holes should be kept to a minimum if the double bottom is transversely framed. If it is longitudinally framed, solid plate floors should be fitted at every frame at least under the main engine.

h.   A symmetrical support at the horizontal centerline of the thrust housing should be adopted.

i.   Particular attention should be given to the design of machinery spaces located in the extreme afterbody, in which the hull stiffness is decreased and part of the structure may be overhanging.

j.   The main-engine seatings should, in general, be integral with the double-bottom structure. In way of the engine foundation, the inner bottom thickness should be substantially increased.

k.   Advantage should be taken of strong longitudinal bulkheads which can constitute the walls of deep-tanks in the forward part of the machinery space.

l.   Special attention should be given to the structural design of machinery spaces when the width at the tank top level is large, and the shape of the space at that level is rectangular or even square. The decreased stiffness due to the large span of transverse members must be taken into account when determining the scantlings and the structural arrangement, particularly in way of the reduction-gear casing.

CHAPTER III   SURVEY OF MAJOR U.S. AND FOREIGN MANUFACTURERS

1.   Geared-Turbine Propulsion Machinery

1.1 General

Problems arising from the incompatibility of hull flexi-
bility and machinery support requirements have been associated
with large vessels having power plants rated 30,000 SHP or more.
The major manufacturers of main propulsion equipment suitable
for such vessels were contacted to determine their requirements
for support of the machinery. In recent years, only three companies
in this country have supplied large geared-turbine units.

Contact with each manufacturer was initiated by a letter
which outlined the background and purposes of the project. It
was suggested that in view of certain unfortunate past experiences
manufacturers in the future will want to scrutinize more care-
fully the environment in which their equipment must function.

Because of the complex hull and foundation structures, in
the past this could only be done qualitatively by visual
examination of the design and by comparison with previous
successful applications using experience and judgement.
Although this procedure worked fairly well for many years, it
had become inadequate as vessel size grew and economic pressures
increased to minimize hull weight. With the aid of computers
and more sophisticated analytical methods, it is now possible
to supplement so-called "eyeball engineering" with quantitative
measures of structural response and thus determine if the
flexibility of the proposed hull design is compatible with
limits established for the deflections of machinery supports.

The manufacturers were requested to:

(a)   define the critical support points for their
      standard machinery designs, and

(b)   indicate the corresponding allowable deflection
      limits.

Four specific areas were suggested for consideration:

(a)   connections between the prime mover and the
      reduction gear,

(b)   internals of the reduction gear,

(c)   main shaft connection to the gear, and

(d)   main thrust bearing.

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Because of the relatively complex nature of the subject, it was felt that direct discussion, in lieu of questionaires, would be desireable and,therefore, meetings were held with the engineering representatives of the manufacturers. Supplementary data were obtained by correspondence and by telephone conversations. A review of the information that was received indicated good general agreement between the three manufacturers and a summary follows.

1.2 Critical Support Points

Fig. 20 illustrates the location of critical points. The table below the figure indicates for each point the directions in which deflections of the supports are significant.

1.3 Connections Between Prime Mover and Gear

Flexible couplings are provided between the turbine rotors and their drive pinions to accomodate relative longitudinal, parallel, and angular movements of these components. Such movements may be caused by thermal growths in the machinery, or by deflections of the supports. Until recently, marine couplings have been of the dental type, each coupling consisting of two gear tooth elements separated by a length of shafting or a sleeve, as shown in simple form in Fig. 21. Each element includes two meshings rings, one having male teeth, the other female teeth.

Gas turbines and recently some steam turbines use another type of coupling in which the teeth of each element are replaced by a flexible diaphragm. Both types may be treated in a similar manner by considering the maximum allowable angular misalignment of each individual element.

Misalignment occurs when the axes of the turbine rotor and pinion are (a) offset but still parallel, and (b) no longer parallel. Angular misalignment of the individual gear tooth elements is found in each case.

The turbines are usually positioned relative to the gear during installation so that under normal, steady-speed operating conditions misalignment of the turbine and pinion axes will be minimized. Changes in operating conditions and deflections of the machinery support points cause misalignment. The amount of misalignment that can be accepted is dependent upon many design factors such as coupling size, torque, rotational speed, tooth design, lubrication, hardness and finish of the teeth, and relative sliding velocity. When a coupling element operates with misalignment,each pair of meshing teeth slide back and forth longitudinally a small amount proportional to the degree of angular misalignment present in the element. The sliding motion is harmonic at a frequency equal to the rotational speed.

LOCATION OF CRITICAL SUPPORT POINTS
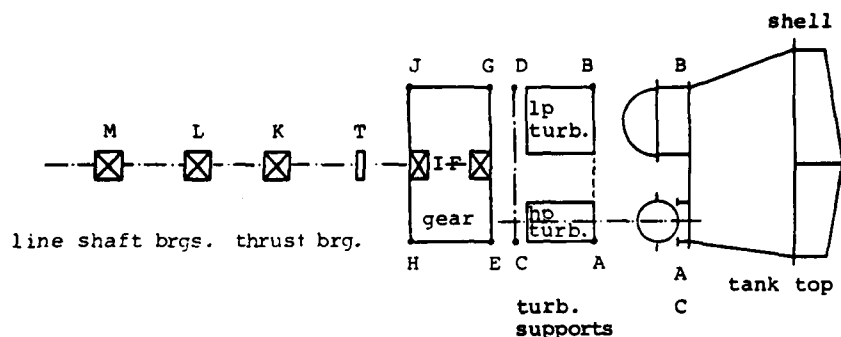
Direction of significant deflections:

Point	Vertical	Athwartship	Longitudinal	Rotational
A	*	*		
B	*	*		
C	*	*		
D	*	*		
E	*	*		
F	*	*		
G	*	*		
H	*	*		
I	*	*		
J	*	*		
K	*	*		
L, etc.	*	*		
T	*	*	*(a)	*(b)

NOTES: (a) important if the arrangement of the foundation permits
a rotational deflection of the thrust bearing housing
upon application of thrust force.

(b) rotation of the thrust housing in a vertical
plane passing through the ship's centerline may cause
the thrust shoes to become unequally loaded and intro-
duce a bending moment in the shafting which will affect
the distribution of load on the low-speed gear bearings.

(c) uniform nonvibratory longitudinal movement of the
gear foundation (points E through J) does not affect the
gear internals if within reasonable limits.

(d) the forward turbine support is assumed to be the
customary plate structure, sufficiently rigid by reason
of vertical depth and stiffening that internal shear
deflection may be ignored. Similar assumptions are
made with respect to the vertical plates forming the
athwartship and longitudinal sides of the reduction
gear foundation.

Figure 20

Research and experience has shown that failures of the
tooth surfaces tend to occur when the maximum sliding velocity
exceeds approximately five inches per second [47]. This criterion
is not sharply defined and, therefore, to provide for a factor
of safety and account for deflection and thermal growths within
the machinery that cannot be avoided, some reduction is required.

Fig. 22 is a diagrammatic representation of a double-
element coupling where the turbine rotor and pinion axes
are parallel but offset by an amount "m", the maximum allowable
parallel displacement. For harmonic motion:

$$m = \frac{60\, v_m L}{RPM\, D}$$

where v_m = maximum allowable sliding velocity
assigned to misalignment caused by
deflection at the supports

L = Spacing between the coupling elements

RPM = rotational speed

D = pitch diameter of coupling teeth

When "m" is evaluated for typical propulsion units of
the size and type under consideration, a value of 0.010
inches is found to be representative. It is convenient to use
the special case of parallel displacement for specification
purposes because of its simplicity.

In the general case where the turbine and pinion axes
are no longer parallel, various alternatives are possible as
shown in Fig. 23. The maximum allowable angular misalignment
"Θ" of an individual element should be obtained from the
manufacturer, or, assuming a properly designed coupling, may
be found from the special case as follows:

$$\Theta = \tan^{-1} \frac{m}{L}$$

In each alternative, the centerline of the spacer has been
extended and the range of allowable angular misalignment of
the turbine rotor and pinion centerlines is "$\pm\, \Theta$".

Fig. 24 represents a typical elevation of turbines, re-
duction gears, and foundations. For illustrative purposes,
assume that the basic hull structure and tank top "XY" bends
to a curve "X'Y'" causing deflections "c", "d", "e", and "f"
at points "C", "D", "E", and "F". These deflections will cause
movements of both turbines and gears which may be assumed as
rigid bodies. If the geometry is known, the angular misalignment
at each coupling may be calculated and compared with the maximum
allowable value "Θ" in order to determine if the deflections
cause the coupling limits to be exceeded.

TYPICAL ARRANGEMENT OF DENTAL TYPE FLEXIBLE COUPLING
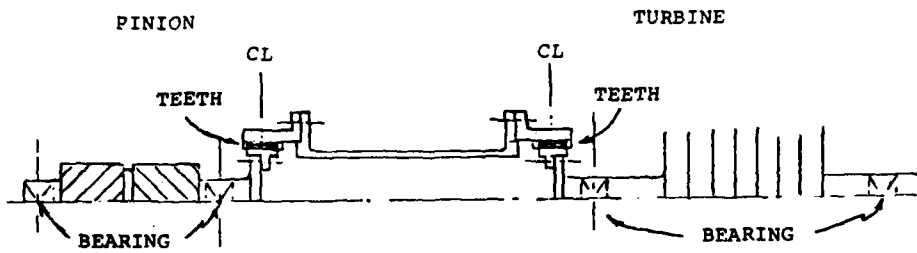
Figure 21

DIAGRAMATIC REPRESENTATION OF A DOUBLE-ELEMENT COUPLING
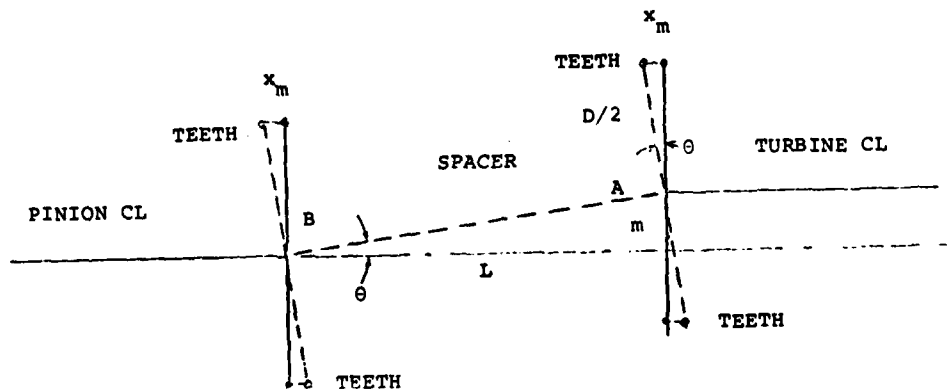
Figure 22

CASES WHERE THE TURBINE AND PINION AXES ARE NOT PARALLEL

DEFLECTION AT "A" RELATIVE TO "B" EQUAL TO "m"
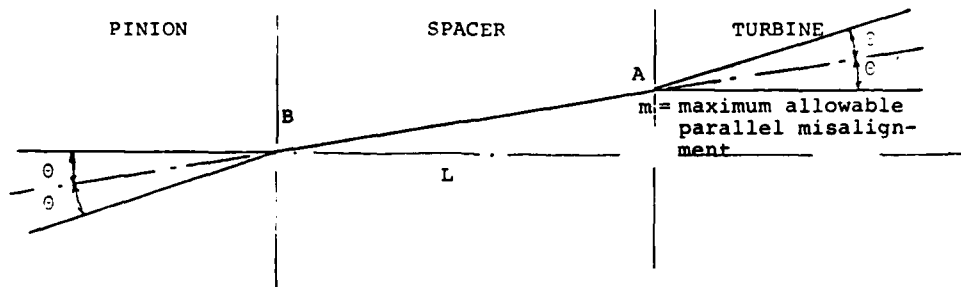
DEFLECTION AT "A" RELATIVE TO "B" LESS THAN "m"
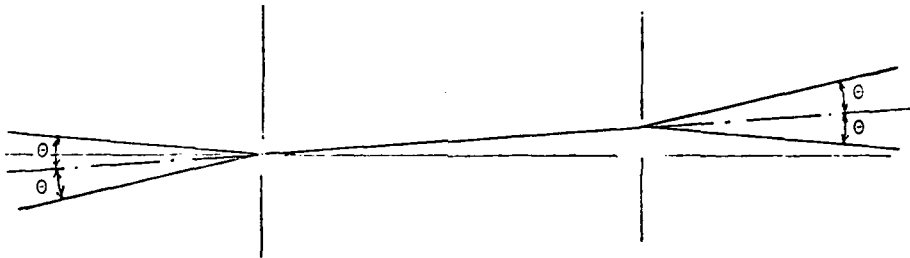

DEFLECTION AT "A" RELATIVE TO "B" GREATER THAN "m"
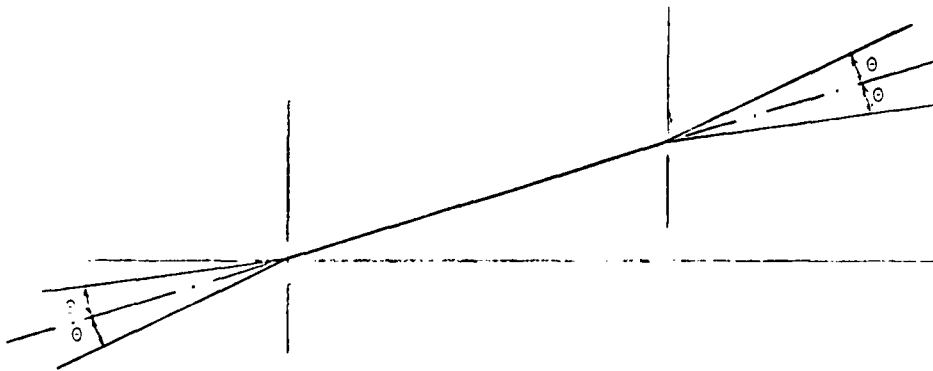
Figure 23

TYPICAL ELEVATION OF TURBINES, REDUCTION GEARS AND FOUNDATION

REDUCTION GEAR   COUPLING   TURBINE
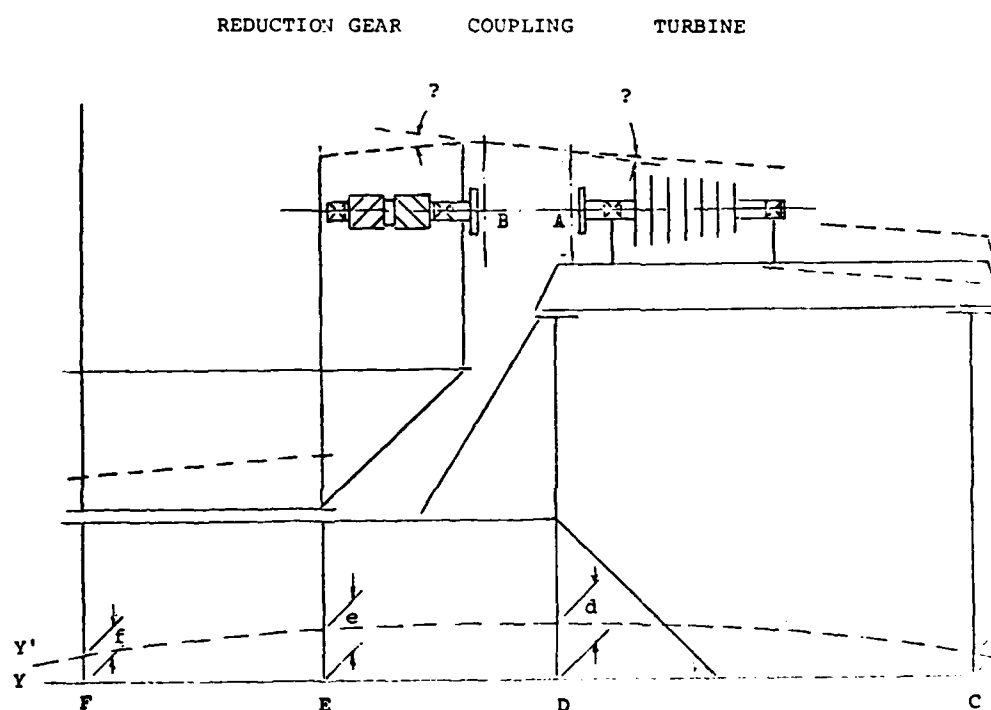
Figure   24

1.4   Internal Alignment of Gears

Most large reduction gears manufactured in this country depend upon the supporting structure to maintain the shape of the gear case and thus preserve the proper alignment of its internal elements.   It is impractical to construct a gear case with sufficient stiffness to successfully resist, or prevent, movements of the foundation.

During manufacture, the seating surfaces of the lower case are machined and maintained in a plane while the internal elements are installed in proper alingnment.   When the unit is placed on shipboard, it is temporarily supported on jack screws and adjusted to restore it to the factory shape.   In addition, tooth contact checks may be made to assure that uniform distribution of load will exist across each mesh under operating conditons.   Minor adjustments of a few mils may be made by raising, or lowering, portions of the case.   Finally, chocks are fitted between the lower case and foundation, usually at least at the four corners of the gear case and in way of the bull-gear bearings.   Thereafter, any deflections of the foundation will be transmitted directly to the gear case and may cause internal misalignment.

Misalignment may consist of either one, or a combination, of two components.   "Difference in center distance" occurs when both pinion and gear axes fall in a plane but the axes are no longer parallel. The result is a variation in depth of teeth engagement along the length of the mesh.   "Out of plane" occurs when the axes no longer fall in a common plane but when viewed normal to the original plane appear to be parallel.   The latter component causes the greater increase in load concentration at the teeth.

Reference [26] published about twenty years ago, stated that the mismatch, or opening, between teeth should not exceed 0.0002 inches per foot of face width.   It appears in the light of present knowledge that this limit is conservative and can be increased to as much as 0.0006 inches per foot.

To relate this limit to allowable deflection at the foundation, it is necessary to consider the position of each pinion.   For example, if the pinion and gear axes fall in a horizontal plane, then vertical deflection at a corner of the foundation will cause maximum "out of plane" misalignment.   In this case, if there is approximately five feet between the second-reduction bearings, the equivalent vertical deflection at a corner of the gear case would be about 0.003 inches.   In large gears of the locked-train type, there are four pinions located in various positions intermediate between horizontal and vertical planes passing through the gear axis and, therefore, both components of misalignment are involved.   The analysis necessary to establish limits in each case can best be performed by the gear

designer, [48]; however, because of the similarity of most modern designs, generalized limitations are possible and are proposed in a following section.

1.5 Main Shaft Connection to the Gear

The bull gear is supported by two large bearings, one forward and one aft. Each journal must be slightly smaller in diameter than its bearing bore in order to provide for lubrication. The operating position of the journals within the clearances are determined by bearing size, oil clearance, rotational speed, and oil viscosity, which in each case is the same for both bearings, and by the magnitude and direction of load on the journals which may differ.

If the design of the bull gear and its shaft were symmetrical and the main shaft disconnected, each bull-gear bearing would be loaded in the same manner. The gear shaft, however, is not symmetrical since it must be extended aft and a flange provided for connection to the main system. The lineshaft, when connected, imposes a direct weight load and introduces a moment at the flange. In this case, the forward and aft bearings are not loaded in the same manner and their journals operate in different positions within the bearing bores. This angular shift of the gear axis with no corresponding movement of the pinions causes misalignment at the meshes and, depending upon the magnitude, danger of tooth wear and failure.

Maximum flexibility in the lineshaft is necessary to minimize effects upon the bull gear. The first lineshaft bearing should be located as far aft of the gear as possible consistent with satisfactory lateral vibration characteristics and bearing loading. In order to achieve additional flexibility, in some cases it may be desirable to select higher strength steel for the line shaft and to reduce its diameter.

"Allowable setting error (ASE)" may be used as a measure of shafting flexibility [29] and is defined as follows:

$$\pm\ ASE = \frac{\Delta R}{I_{11} - I_{22}}$$

where ΔR = allowable difference between two slow-speed gear bearing vertical static reactions

I_{11} = reaction influence number of forward slow-speed gear bearing on itself

I_{22} = reaction influence number of aft slow-speed gear bearing on itself

ASE represents the amount of vertical parallel movement
that the gear may be raised, or lowered, relative to the line-
shaft bearings without exceeding ΔR. ΔR is established by the
manufacturer and may vary from approximately 10,000 - 20,000
pounds for large locked-train units. Reference [29] recommends
an absolute minimum ASE value of $\pm$ 0.010 inches, but this is
based on experience with smaller vessels with greater hull
rigidity. Recent experience indicates that ASE values of 0.025
inches are practical and values less than 0.020 inches should
be avoided, particularly in larger hulls. ASE may be used for
comparative purposes but the acceptability of the shafting
design should be based upon a complete evaluation of the
differences in both horizontal and vertical static reactions
at the forward and aft gear bearings relative to limits set
by the manufacturer.

1.6 Main Thrust Bearing

The arrangement, location, and other consideration associa-
ted with the main thrust bearing are dealt with in Chapter II.
No additional specifics were developed during the survey.

1.7 Criteria

A review of the information gathered during the survey has led
to proposed criteria which is for convenience shown on the next page.

1.8 Alternative Types of Geared-Turbine Propulsion Machinery

Practically all large-reduction gears for turbine drives
manufactured in this country during recent years have been
of the locked-train design; however, several manufacturers
commented on other types including several which incorporated
features believed to render them less susceptible to the
effects of foundation movements. This increased capability
to successfully handle deflections of the supports unfortunately
is accompanied by disadvantages.

The alternatives cost more to manufacture and, therefore,
it is unlikely that a manufacturer would offer a more expensive
unit considering the highly competitive nature of the marine
propulsion market unless it was specifically required by bidding
specifications. Some types involve major rearrangements of
machinery spaces requiring more floor area and longer length.
There may be some loss of accessibility for maintenance and
repair depending upon the machinery space arrangement. Finally,
for each manufacturer there may be development costs and risks
associated with a new design.

Before considering the use of such alternatives, the
flexibility of the proposed hull should be investigated at
an early point in the design schedule to determine if the

PROPOSED DEFLECTION LIMITS FOR FOUNDATIONS

OF

GEARED TURBINE PROPULSION UNITS

The main turbines, reduction gears and condensers shall be designed and coupled to each other so that the deflections of supporting ships structure caused by combinations of elastic and thermal effects resulting from seaway, torque and thrust reactions, cargo and ballast loading, etc., under any operating condition shall not produce excessive stresses or wear, provided these deflections do not exceed the following:

1. Vertical movement of 0.004 inches at the seating surface at the forward port (starboard) corner of the reduction gear foundation relative to a plane passing through the following three points in the seating surface -

   (a) the aft port (starboard) corner

   (b) the forward low-speed gear bearing

   (c) the aft low-speed gear bearing

2. Relative movements of line shaft bearings and gear foundation which cause the difference in static vertical reactions at the bull-gear bearings to exceed the maximum allowable value (ΔR) established by gear manufacturer.

3. Relative movements of line-shaft bearings and gear foundation which cause the difference in static horizontal reactions at the bull gear bearings to exceed the maximum allowable value established by the gear manufacturer.

4. Parallel displacement of 0.010 inches, or equivalent angular misalignment, of a turbine rotor to pinion.

movements of the machinery supports under normal operating conditions fall within allowable limits. Should these limits be exceeded, consideration may be given to additional stiffening in the foundations or basic hull structure. In the event that this is impractical, or uneconomical, to provide sufficient stiffness, then the use of alternative machinery types may be warranted and, if so, should be specified after consulting with all potential machinery suppliers.

The articulated double-reduction gear design permits a greater variety of arrangements with essentially the same rotating parts by rolling the pinion and gear centers to various positions and may be classified by the number of horizontal planes in which the pinion and gear axes fall. The "three plane unit" [(1) turbines and high-speed pinions, (2) high-speed gears and low-speed pinions, (3) low-speed gear] is the most common [27]. The gear case is a common structure serving both first and second reductions. The lower gear case usually is chocked and bolted to the foundation at the corners, athwartship wall between first and second reductions, low-speed bearings and often at other points. Deflections of the foundations are transmitted directly to the gear case and may cause misalignment of the internals

"Singles" and "two plane" arrangements have been built with separate and independent gear cases for the first and second reductions. Flexible couplings between the two reductions allow for relative movements between the casings. The low-speed pinion and low-speed gear axes fall in the same horizontal plane. Such an arrangement is less sensitive to forces and moments introduced by the line shafting. Misalignment of the pinion and gear axes in the horizontal plane (in-plane misalignment) causes a variation in depth of tooth engagement and is less disturbing to the uniformity of tooth contact across the face of each helix. Horizontal misalignment may be caused by differences in horizontal forces and by differences in vertical forces on the low-speed gear bearings.

"Out-of-plane" misalignment is more serious because it results in a greater load concentration at the mesh. It can be eliminated, or minimized, when caused by foundation movements through the use of "three point chocking". Consider each half (port and starboard) of the lower gear case and the corresponding mesh between pinion and gear. Provide chocks at the midpoint of the longitudinal side and in way of the two low-speed bearings. These three support points will always remain in a plane regardless of deflection at any, or all, of these points, and, therefore the alignment of pinion and gear will not be affected. This arrangement leaves the corner of the gear case unsupported and requires careful attention to the design of the case. One European manufacturer has found it desirable to place springs at the corners. In the case of single-plane gears having

separate first and second-reduction gear cases, it is possible
to treat the first-reduction gear cases in a similar manner.

Geared-turbine propulsion units for naval service have
been built on platforms which provide sufficient stiffness
and support for the components. The platform is floated on
springs, or isolation mounts, which essentially eliminate any
effects of deflections at the support points. The principal
purpose, however, is to reduce noise transmission to the hull
and thus avoid detection by sonar. This arrangement does not
appear to be practical for application to large merchant type
units. It is costly, not only because of the additional struc-
tures but also because the drive and all piping connections
must be flexible. Further, the output torque of a large
merchant unit is from two to ten times greater than naval
units where this arrangement has been applied.

A naval unit that is platform mounted and spring supported
requires a flexible coupling between the gear and line shaft
to accomodate relative movements. Merchant type units are
usually solidly bolted to the lineshaft; however, consideration
is currently being given to large flexible couplings designed
to handle the much greater torque found with high-powered
merchant units. The use of flexible couplings would eliminate
the introduction of bending moments at the gear shaft flange
leaving only weight loads to be considered.

It is interesting to note that in the early history of
marine gears when the current high-accuracy gear cutter were
not available, various devices were adopted in attempts to
compensate. One arrangement involved a "floating pinion"
with bearings that were spring supported. This device, and
others of a similar nature, became obsolete as gear cutting
became more accurate; however, it should be remembered that
at that time ships were smaller and foundations were more
rigid.

2. Diesel Engines

2.1 General

Diesel marine propulsion systems are receiving now a
great deal of attention. As mentioned in Chapter I in the
case of ships equipped with diesel propulsion plants, problems
of hull-machinery incompatibility have also been found in the
past. With the decrease in power of diesel engines, the dimen-
sions of the webs, crankpins and journals of crankshafts have
increased considerably, and as a result the overall crankshaft
stiffness has increased. This fact, coupled with an increase
in flexibility of hull girder, double bottom and bedplates of
propulsion plants, has been the cause of crankshaft and
bearing damage.

In an extensive experimental study described in [49] and involving 32 ships, it was found that deformations of the bedplates due to cargo and the effect of sea loads were quite noticeable. The amplitudes measured were of the order of 0.3 to 2 mm in the case of cargo loads, and of the order of 1 to 7 mm in the case of sea loads. These amplitudes were measured relative to the bedplate deformed profile in the light condition and in calm water. This indicates that the vertical positions of the crankshaft bearings vary considerably, according to the loading conditions and sea state. Thermal effects, such as temperature differentials between the cylinder block and its supporting base, can also be responsible for considerable bedplate deformation, and these can induce considerable loads even on the engine structure itself.

The study of the rigidity of the crankshaft and its components is difficult. In [50], a conventional strength of materials approach is adopted, which leads to the distribution of vertical and horizontal reactions in way of each bearing, to the influence coefficients of each bearing of the crankshaft assembly, and to the openings/closings of the webs. In carrying a structural analysis of machinery components, a more exact definition of the structure's geometry can be obtained by using the finite-element method, as suggested in [51].

It was found, as discussed in [49] that when the engine is stationary, and the main bearings are lined in a straight line, because of the fact that the different crankshaft sections have different rigidities, the reactions at the main bearings are unequal. When the engine is running, dynamic and thermal effects can even cause the reactions at some bearings to be reversed, so that as a result, the journals may lose contact with the lower shells.

The inherent flexibility of the crankshaft and the related failures, such as the loss of contact between some of its journals and the corresponding bearings, show how important it is to measure crankshaft deflections and check if they fall within reasonable limits. Part of the problems due to excessive flexibility can be reduced if a curved crankshaft alignment procedure is adopted, as suggested in [49].

An important factor in the hull-engine foundation compatibility is the design of the engine structure itself. In conventional diesel-engine design, the structure of the crankcase is essentially composed of columns extending between the bedplate and the cylinder block. This particular arrangement implies a relatively low shear stiffness, and since large two-stroke engines essentially behave as beams in pure shear, the structure is very much affected by the degree of deformation

of the hull and engine-support systems. A possible alternative
arrangement consists of box-shaped longitudinal girders com-
prising a deep-section single-walled bedplate and high cast
cylinder jackets [8,24]. As a result of the large bending and
shear stiffness of box girder construction, it has been possible
to decrease substantially the deformations induced by the ship's
girder and double bottom, and no abrupt changes in curvature
are observed. Thus the crankshaft line bearing is deformed to
a smaller degree, increasing the likelihood of a good engine
performance. Numerical studies using the finite-element
method and model tests have indicated that this type of arrange-
ment can lead to an increase of about 50 to 80 percent in the
standard hull foundation structural stiffness, as compared
to an increase of approximately 20 to 30 percent in the case
of column type of design [24]. Thus, the double-bottom deforma-
tion which occurs in normal operation can be absorbed by an
engine of this kind. Furthermore, with large two-stroke engines,
the maximum engine flexure produced from double bottom deforma-
tion rarely exceeds 1mm. Thus, this indicates that with modern
diesel-engine designs, if the structural arrangement of the
engine spaces is carefully planned, the problem of hull-
foundation rigidity compatibility can to a large extent be
minimized.

2.2 Manufacturers' Requirements

Engine manufacturers provide shipyards with recommended
foundation structural arrangements for given engines, in order
to ensure that the reaction forces between bedplate and founda-
tion are properly transmitted.

Typically, these include recommendations on scantlings
for top plate, bottom shell plating and longitudinal members,
design and arrangement of lube oil service tank, chocks and
foundation bolts. The design of the lube oil service tank
is important in order to allow a uniform movement of the
whole engine due to thermal expansion. It appears that often
the scantlings recommendations provided by the manufacturer are
too general, since they don't take into account the ship's
longitudinal bending or local deflections within the engine
spaces. Thus, the naval architect must use his own judgement
in designing the structure in way of the engine spaces. The
structural design procedures presented in this report (section IV,
Proposed Method) could help the designer to form a basis for
judgement. The recommended bolting and their bearing arrangements
are, on the other hand, usually more specific, and should be
followed.

When defining the deformation limits for the engine founda-
tion, the most important deformation mode to be considered is
bending in a vertical plane containing the longitudinal axis
of the engine. The deflection profile due to the double bottom
deformation in way of the engine closely approximates a circular
arc, and a general guideline for large two-stroke engines is

that between extreme ship load conditions the radius of curvature of this circular arc should not be less than 30×10^3m.

Engine manufacturers have used for many years the measurement of the crankshaft web deflection as the criterion for the alignment condition of the engine. The crankweb deflection is the difference of the distance between two webs when the crankshaft has been rotated through 180°. The measurement can easily be carried out with a special crankshaft alignment gauge or chocking device.

The web deflection leads to important information regarding the crankshaft stress, the bearing load and the quality of the engine chocking. Taking into account these criteria and based on past experience, the admissible values for the web deflection of each engine can be specified. Values are usually given for two different cases: the case in which the engine is still new and the case in which the engine has been in operation for some time, so that some wear of bearings or chocks has taken place. In the first case, the permissible web deflections can be considerably lower. If in any case, the values measured exceed the corresponding maximum permissible levels the cause must obviously be eliminated and the crankshaft must be realigned. Possible causes of excessive deflection are an uneven wear of the crankshaft bearings, a change in the position of the driven shaft or a change in the support conditions for the engine. It is obviously important that each time the crankweb deflections are measured the loading and thermal conditions of the ship be essentially the same.

Given the allowable crankweb deflections, it is possible to determine the overall admissible crankshaft deflections, using either conventional strength of materials approaches [50], simple graphical methods or more sophisticated finite-element methods [51]. If the permissible crankshaft deflections are known, the maximum support structure distortions can also be defined. As indicated in [8], the use of modern computational techniques for the structural analysis of the entire engine now enables the diesel manufacturers to specify qualitative data on the acceptable deformation for engines under various load conditions.

Due to the variety of diesel engines presently available, no general foundation rigidity criteria will be presented here. This should be supplied for each case by the manufacturer. The naval architect can then essentially use the method suggested in this study in order to make sure that these requirements are met. In particular, a good structural arrangement of the engine spaces, following the general guidelines set forth in Chapter II should always be adopted, as a first step towards minimizing the possibility of hull-machinery incompatibility.

CHAPTER IV.   PROPOSED METHODOLOGY

1.   Methods for Evaluating the Foundation Stiffness

The methods for evaluating the machinery foundation stiffness are for convenience subdivided here into the following two groups:

- stress-hierarchy method (SHM)

- finite-element method (FEM)

In the stress-hierarchy method [19], the structure is represented by different models with an increased degree of refinement, starting with a beam model for the hull girder, and using two and three-dimensional frame models or grillage models in order to represent the transverse framing system and the double bottom in way of the engine room. In the finite-element method, the structure is discretized in detail so that the actual geometry and configuration can be accurately represented. The advantage of the stress-hierarchy method (SHM) is the capability it offers to the ship structural designer to determine at the early design stages the relative merit of the various possible ways of supporting the machinery components at a reasonable computer cost. Once the design has gone into its more detailed phase, then a FEM is needed in order to get a better estimate of resulting deflections. Clearly the results of the SHM are of great help in reducing the number of computer runs for the FEM calculations. The various methods for evaluating the foundation stiffness, as listed in Table IX, will now be discussed.

2.   Stress Hierarchy Method

The total engine structural deflection can be separated into the following components:

- deflections due to hull girder deformation

- deflections due to transverse web frame deformation

- deflections due to engine-room double-bottom deformation

2.1 Deflections Due to Hull Girder Deformation

The hull-girder is considered as a free-free beam subjected to laterally and axially applied loads. The lateral loads are due to the buoyancy and hull weight forces. The axial loads are due to the application of the thrust force at the height of the shaft centerline which, in general, does not

TABLE IX

METHODS FOR EVALUATING FOUNDATION STIFFNESS

Stress-Hierarchy Method (SHM)

- Hull Girder Deformations:

    a. Bernoulli Beam (no shear effects)

    b. Timoshenko Beam (shear effects included)

- Transverse-Frame Deformations

    a. 2D Frame Model

- Double Bottom Deformations:

    a. Beam Model for Floors

    b. Grillage Model

Finite-Element Method (FEM)

- Coarse Mesh

- Fine Mesh

coincide with the neutral axis of the hull-girder thus introducing externally applied moments. The total hull girder deflection can be separated into the following components:

- deflections due to bending deformation

- deflections due to shear deformation

2.1.1 Hull Girder Deflections Due to Bending Deformation

The distributed load q per unit length due to buoyancy and weight forces is

$$q = b-w \qquad (1)$$

where b is the buoyancy force per unit length and w is the ship weight per unit length. Forces are considered positive when directed upwards. The shear force V at a location x is obtained by integrating equation (1)

$$V = \int_{0}^{x} (b-w)\,d\xi \qquad (2)$$

where ξ is an integration variable and x is the hull-girder neutral axis which is oriented from the aft to the forward end and originates at the aft end. Integrating the shear force distribution, the bending moment M at location x can be obtained as follows:

$$M = \int_{0}^{x} \int_{0}^{\zeta} (b-2)\,d\xi d\zeta + M^{C} < x - a >^{O} \qquad (3)$$

where ξ and ζ are integration variables, $< >^{O}$ is the Dirac delta function, and a is the distance of the applied concentrated moment of magnitude M^{C} from the aft end.

The positive direction for the shear force V and bending moment M is given in Fig. 25

SIGN CONVENTION

Fi e 25
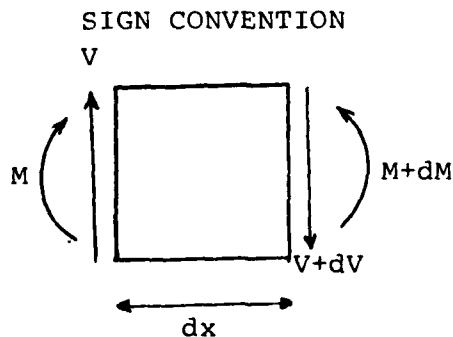
The slope Θ of the deflection curve at location x is

$$\Theta = \int_0^x \frac{M}{EI}\, d\xi - \int_0^L \frac{M}{EI}\, d\xi \tag{4}$$

where EI is the hull girder flexural rigidity, and L is the length of the hull girder. The positive sense for the slope is in the clockwise direction. The hull girder transverse deflection $(y_B)_{HG}$ due to bending deformation is

$$(y_B)_{HG} = \int_0^x \int_0^\xi \frac{M}{EI}\, d\xi d\xi - \frac{x}{L} \int_0^L \int_0^\xi \frac{M}{EI}\, d\xi d\xi \tag{5}$$

Deflections are considered positive when directed upwards.

2.1.2. <u>Hull Girder Deflections Due to Shear Deformation</u>

The hull girder transverse deflection $(y_S)_{HG}$ due to shearing deformation is

$$(y_S)_{HG} = f\left\{ \int_0^x \frac{V}{GA_w}\, dx - \frac{x}{L} \int_0^L \frac{V}{GA_w}\, dx \right\} \tag{6}$$

where A_w is the total area of the web (comprising the area of all vertical parts of the shell, longitudinal bulkheads and girders), G is the shear modulus and f is a form factor (equal to 1.2 for rectangular cross-sections).

2.1.3. <u>Calculation Procedure</u>

The total weight w per unit length is essentially given by

$$w = w_{HS} + w_C + w_M \tag{7}$$

where w_{HS} is the hull steel weight per unit length, w_C is the cargo weight per unit length and w_M is the machinery weight per unit length. For tankers the hull weight distribution depicted in Figure 26 can be adopted.

HULL WEIGHT DISTRIBUTION FOR TANKERS
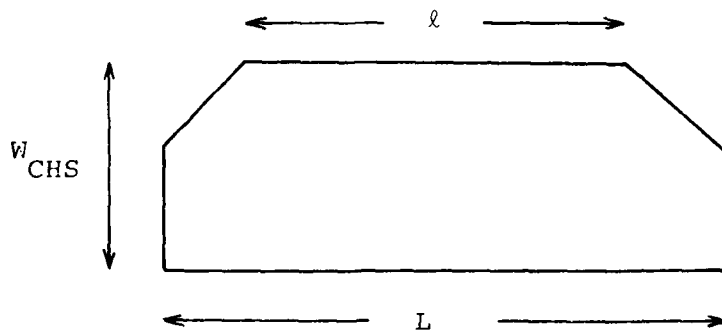
Figure 26

The weight per unit length in the region of the parallel middle body is [52, 53]

$$w_{CHS} = \frac{w_{HS}}{4L} \left(5 - \frac{\ell}{L}\right) \qquad (8)$$

where ℓ is the length of the parallel middle body. Given the hull steel weight per unit length w_{HS} and the location of the longitudinal center of gravity L_{CG} the determination of the weight distribution can be completed. The area under the weight curve is determined for each loading condition using the trapezoidal rule of integration.

The area under the buoyancy curve is determined by parabolic interpolation between input data points of successive stations. Performing the integrations indicated by equations (2) through (6) the shear force, the bending moment, the slope of the deflection curve and the deflection due to bending and shearing deformations are respectively obtained.

2.2   Deflection Due to Transverse Web Frame Deformation

2.2.1   Symmetric Response

The symmetric response of a transverse web frame due to
hydrostatic pressure is sought herein (Fig. 27).

TRANSVERSE WEB FRAME
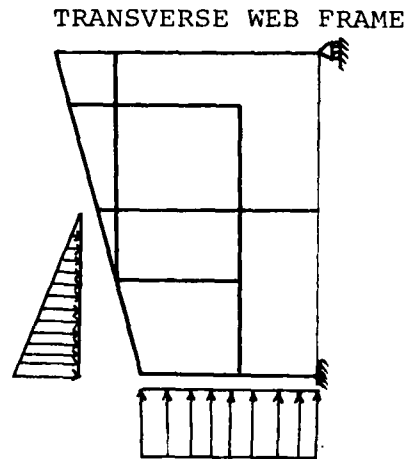
Figure 27

The web frame is symmetrical with respect to the vertical
mid-plane. The loading is also symmetrical about the same
plane. Therefore, material particles in the plane of symmetry
remain in the plane with rotations possible only about the
normal to the plane.

The Node Method for plane frames [54] is utilized to
determine the required response. The formulation of the
node method is based on the following two basic assumptions:

- the plane frame is loaded only at its nodes

- the plane frame has completely "fixed" supports
  (i.e. at a support both the displacement vector
  and the rotation are assumed to be zero).

The first assumption requires a method for reducing to nodal
loads any externally applied distributed and/or intermediate
loads. This is accomplished by decomposing the given loading
into the forces required for zero nodal point displacements
(fixed-end quantities) and the nodal forces used in the
analysis which are negative of the fixed-end quantitites.
To obtain the true state of stress in a member having dis-
tributed loading, the results obtained from both parts must
be superimposed.

The second assumption which requires the use of fixed supports is not a restriction on the range of applicability of the method. Thus, for instance, a frame with hinged support is equivalent to a frame with fixed support with member release.

The nodal displacement vector $\underset{\sim}{\delta}$ and the applied force vector $\underset{\sim}{P}$ is defined for the entire structure:

$$\underset{\sim}{\delta}^T = [\underset{\sim}{\delta}^T_1, \ \underset{\sim}{\delta}^T_2, \ldots \underset{\sim}{\delta}^T_J] \text{ and } P^T = [\underset{\sim}{P}^T_1, \ \underset{\sim}{P}^T_2, \ldots \underset{\sim}{P}^T_J] \qquad (9a,b)$$

where $\underset{\sim}{\delta}^T_i = [\delta_{ix}, \ \delta_{iy}, \ 0_i]$ and $\underset{\sim}{P}^T_i = [P_{ix}, \ P_{iy}, \ M_i]$ (10a,b)

x,y are the global cartesian coordinate axes, δ_{ix}, δ_{iy} are the components of the displacement vector along the x and y axes respectively at node i, 0_i is the rotation at node i, P_{ix}, P_{iy} are the x and y components of the applied force vector at node i, M_i is the applied moment at node i, the symbol $(\sim)$ denotes a vector quantity, the symbol $(\sim)^T$ denotes the transpose of a vector quantity and J is the number of movable joints.

The member displacement vector $\underset{\sim}{\Delta}$ and the member force vector F is defined for the entire structure:

$$\underset{\sim}{\Delta}^T = [\underset{\sim}{\Delta}^T, \underset{\sim}{\Delta}^T, \ldots, \underset{\sim}{\Delta}^T_B] \text{ and}$$

$$\underset{\sim}{F}^T = [\underset{\sim}{F}^T_1, \ \underset{\sim}{F}^T_2, \ldots, \underset{\sim}{F}^T_B] \qquad (11a,b)$$

where $\underset{\sim}{\Delta}^T_i = [\Delta L_i, \ \alpha_i^+, \alpha_i^-]$ and $\underset{\sim}{F}^T_i = [t_i, \ m_i^+, m_i^-]$ (12a,b)

The local coordinate system of the ith member consists of a cartesian coordinate system x^i, y^i where x^i is in line with the neutral axis of the member and directed from the negative end towards the positive end (Fig. 28).

LOCAL COORDINATE SYSTEM
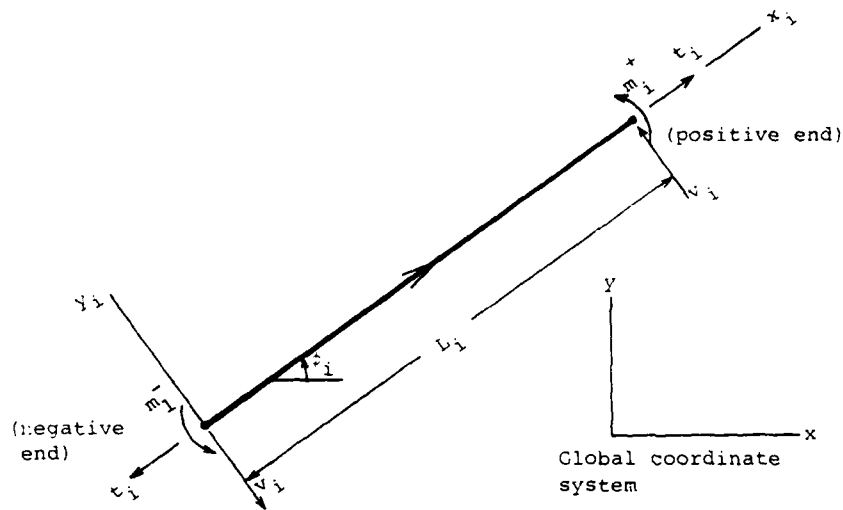
Figure 28

The member bending moment, shear force and axial force at the positive end are m_i^+, v_i, t_i respectively. Corresponding quantities are defined for the negative end. The length change of the ith member, the rotation of the positive end less the rigid body rotation of the member and the rotation of the negative end less the rigid body rotation of the member are ΔL_i, α_i^+ and α_i^- respectively.

For a plane frame structure composed of uniform straight beams, the stiffness matrix $\underset{\sim}{K}^L$ for the entire structure is:

$$
\underset{\sim}{K}^L = \begin{bmatrix} \underset{\sim}{K}_1 & & \\ & \underset{\sim}{K}_2 & \underset{\sim}{0} \\ & & \ddots \\ \underset{\sim}{0} & & \underset{\sim}{K}_E \end{bmatrix} \text{ where } \underset{\sim}{K}_i = E \begin{bmatrix} A_i/L_i & 0 & 0 \\ 0 & 4I_i/L_i & 2I_i/L_i \\ 0 & 2I_i/L_i & 4I_i/L_i \end{bmatrix}
$$

where $\underset{\sim}{K}_i$ is the stiffness matrix for the ith member, expressed in its local coordinate system A_i, L_i and I_i are the cross sectional area, the length and the moment of inertia of the ith member, respectively, and E is Young's modulus. The stiffness $\underset{\sim}{K}^G$ expressed in the global coordinate system is

$$
\underset{\sim}{K}^G = \underset{\sim}{N}^T \underset{\sim}{K}^L \underset{\sim}{N}
$$

where $\underset{\sim}{N}^T = [\underset{\sim}{N}_1, \underset{\sim}{N}_2, \ldots\ldots, \underset{\sim}{N}_B]$

$$
\underset{\sim}{N}_{ij} = \begin{cases} \underset{\sim}{N}_i{}^+ \ \underset{\sim}{R}_i & \text{if j is the positive end of member i} \\[2mm] \underset{\sim}{N}_i{}^- \ \underset{\sim}{R}_i & \text{if j is the negative end of member i} \\[2mm] O & \text{otherwise} \end{cases}
$$

$$\text{(14c)}$$

$$
\underset{\sim}{N}_i{}^+ = \begin{bmatrix} 1 & O & O \\ O & -L_i{}^{-1} & -L_i{}^{-1} \\ O & 1 & O \end{bmatrix} \quad \text{and} \quad \underset{\sim}{N}_i{}^- = \begin{bmatrix} -1 & O & O \\ O & L_i{}^{-1} & L_i{}^{-1} \\ O & O & 1 \end{bmatrix}
$$

$$\text{(14d,e)}$$

$$
\underset{\sim}{R}_i = \begin{bmatrix} \cos\phi_i & \sin\phi_i & O \\ -\sin\phi_i & \cos\phi_i & O \\ O & O & 1 \end{bmatrix}
$$

$$\text{(14f)}$$

$\underset{\sim}{R}_i$ is the rotation matrix for the ith member and ϕ_i is the inclination angle ϕ_i for the ith member in the global coordinate system (Fig. 28).

The modal displacement vector $\underset{\sim}{\delta}$ and the applied force vector $\underset{\sim}{P}$ are related through

$$
\underset{\sim}{P} = \underset{\sim}{K}^G \ \underset{\sim}{\delta}
$$

$$\text{(15)}$$

The problem is solved by constructing the stiffness matrix $\underset{\sim}{K}^G$ using equations (14) and then inverting equation (15) to obtain

$$
\underset{\sim}{\delta} = (\underset{\sim}{K}^G)^{-1} \underset{\sim}{P}
$$
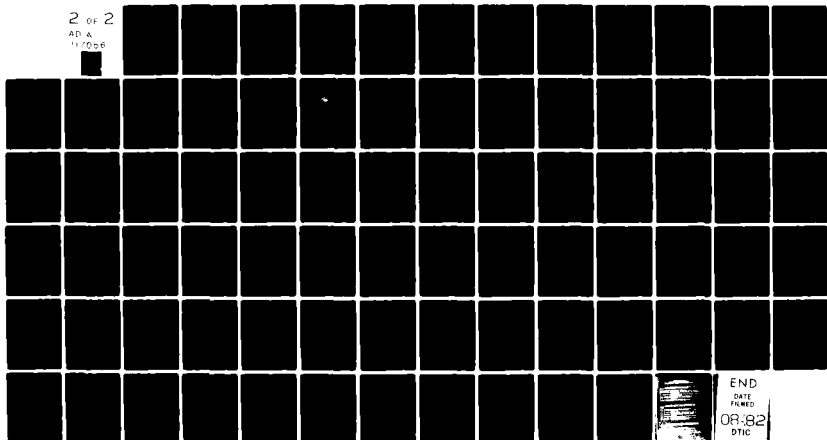
2.2.2 Antisymmetric Response

When the ship moves obliquely across a wave system, then
the slope of the wave changes at each section of the ship,
which means that the draft of water on one side is different
from that on the other side at a particular section. Thus a
horizontal force is generated at each section since the pressures
on the two sides of the ship are different. The sign of the hori-
zontal force changes along the length of the ship so that bending
in the horizontal plane results.

The web frame is symmetrical with respect to the vertical
mid-plane. The hydrostatic pressure loading due to the oblique
wave incidence can be decomposed into components symmetrical
and antisymmetrical with respect to the vertical plane of symmetry.
In the case of the antisymmetric loading, the frame undergoes
deformations antisymmetrical with respect to the plane of
symmetry of the structure, and the material particles in the
plane of symmetry leave the plane along its normal direction
with rotations possible only about the lines parallel to the
plane. Thus, in this case the boundary conditions are as
depicted in Fig. 29.

BOUNDARY CONDITIONS
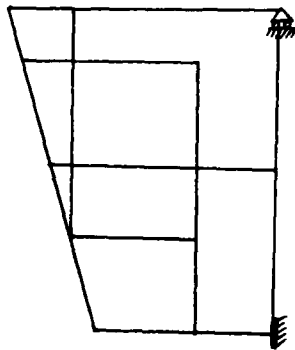
Figure 29

The structural response procedure is similar to the approach
outlined to obtain the symmetric response. What remains to be
determined is the magnitude of the pressure loading due to the
oblique wave incidence.

The following assumptions are made:

- the breadth of the ship to the wavelength ratio
  is assumed to be small in comparison to one

- no allowance is made for the orbital motion of the particles in the wave, i.e., the problem is treated as a purely static one

- torsion is not considered, i.e. both due to vertical and horizontal pressures

- the decoupled problem is treated, i.e. the horizontal bending problem is considered separately from the vertical bending problem.

The direction of vessel advance has an angle α with the direction of wave propagation (Fig. 30).

HULL POSITION RELATIVE TO WAVES
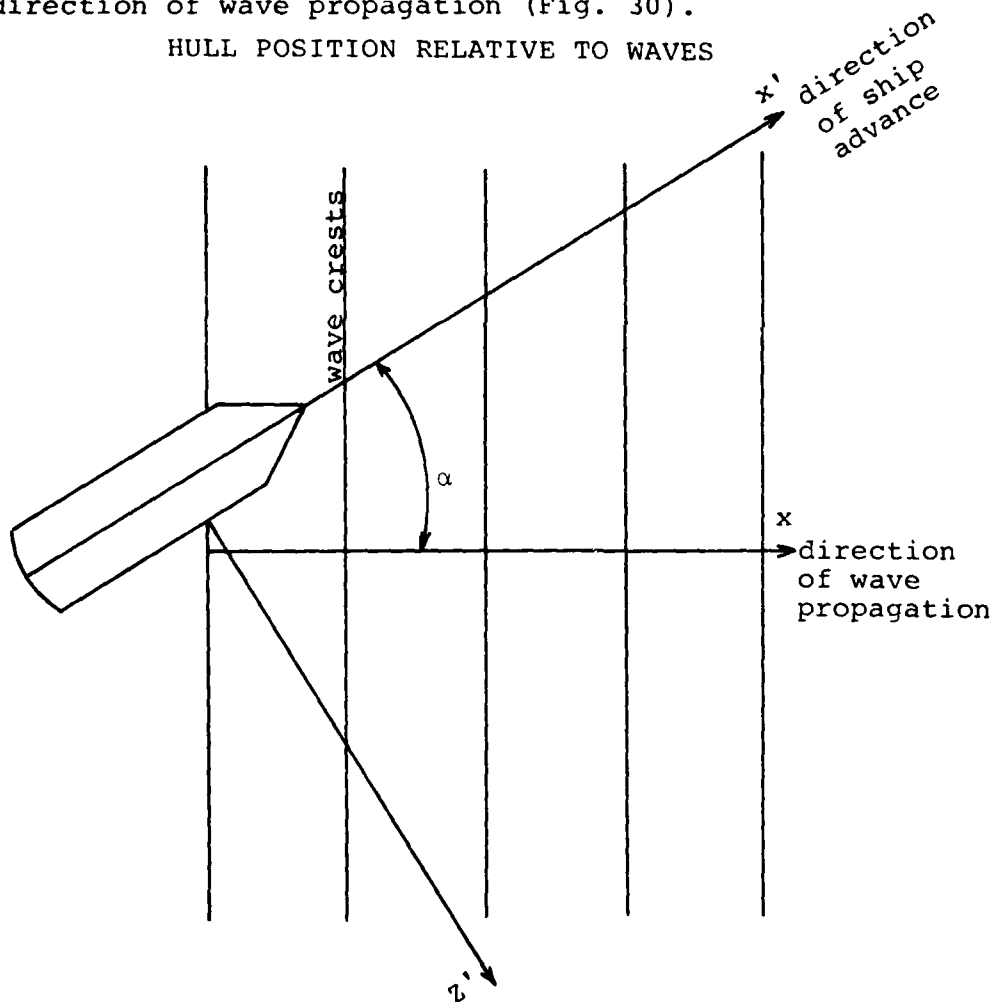
Figure 30

The coordinates x, x' and z' are related through

$$x = x'\cos\alpha + z'\sin\alpha \tag{17}$$

The wave surface elevation η_w (considered to be positive when upwards) is

$$\eta_w = A\cos\left(\frac{2\pi x}{1} - \omega t\right) \tag{18}$$

where   A is the wave height

1 is the wave length

ω is the wave frequency

t is the time.

Substituting the equation (17) into (18)

$$\eta_w = A\cos\left(\frac{2\pi x'}{1}\cos\alpha + \frac{2\pi z'}{\ell}\sin\alpha - \omega t\right) \tag{19}$$

The wave slope for z' = 0 and t = 0 is

$$\frac{\partial \eta_w}{\partial z'} = A\,\frac{2\pi}{1}\,\sin\left(\frac{2\pi x'}{1}\cos\alpha\right)\sin\alpha \tag{20}$$

The ordinates of the pressure loading d_1, d_2 (Fig. 31) due to oblique wave incidence are:

$$d_1 = T - \frac{B}{2}\left(\frac{\partial \eta_w}{\partial z'}\right) \tag{21a}$$

and $$d_2 = T + \frac{B}{2}\left(\frac{\partial \eta_w}{\partial z'}\right)$$

where   T is the ship draft

and   B is the ship breadth.

POSITIVE WAVE SLOPE
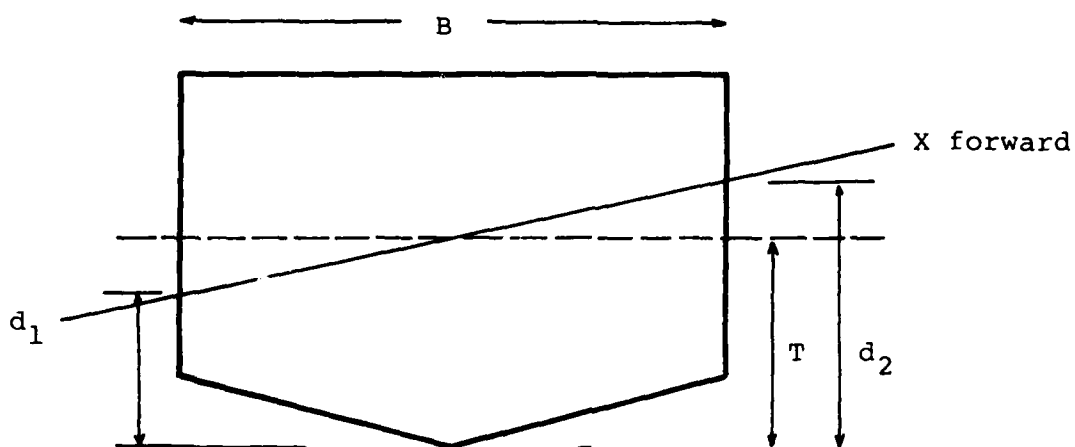
Figure 31

2.3 Deflections Due to Engine Room Double-Bottom Deformation

2.3.1 Grillage Method of Analysis

The engine room double-bottom foundation structure is considered as a grillage consisting of transverse floors and longitudinal girders. Both bending and shear deformations are retained in the analysis. Clamped boundary conditions are considered along the aft and forward engine room transverse bulkhead and the outer shell. The applied loading in the present case consists of the following components:

- hydrostatic pressure

- machinery deadweight loading

- thrust force

The method of approach for solving such a grillage problem is very similar to the one described in section 2.2. In fact the method considered for plane frames can be easily generalized to address three-dimensional frames. The grillage problem is then a particular case of the three-dimensional frame problem. Thus, both deflections, due to transverse web frame deformation and due to engine room double-bottom deformation, can be calculated by using a computer code for space frame problems.

2.3.2 Beam Method of Analysis

A more simplified model results by considering the transverse floor in question as a Bernoulli beam. In such a case, the transverse floor deflections w_{TF} (in the transverse plane) are given by the solution of the following differential equation:

$$\frac{d^4 w_{TF}}{dx^4} = \frac{\rho_w g T s_{TF}}{(EI)_{TF}}$$

where x is the transverse floor beam longitudinal axis $(EI)_{TF}$ is the transverse floor flexural rigidity, assumed constant

ρ_w is the water mass density

T is the ship's draft

and s_{TF} is the transverse floor spacing.

In addition to the transverse floor deflections w_{TF}, the deflections of the engine room double bottom in the vertical plane must also be considered. These deflections can also be estimated using a suitable equivalent beam model [10].

The various components of the stress hierarchy method just described are summarized for convenience in Table X.

3. Finite-Element Method

The Finite-Element Method (FEM) is now a widely adopted tool in ship structural analysis, and it has already been used quite extensively in ship hull-machinery compatibility studies, e.g. [2,5,7,17,19,51]. Due to the very small order of magnitude for deflections which are of concern here, i.e. thousandths

TABLE X

STRESS HIERARCHY METHOD

Hull Deflections

Timoshenko beam (shear deformations included)

Hull Girder Deformations

Bernoulli beam (shear deformations not included)

$$\frac{d^4 w_{HG}}{dx^4} = \frac{?}{(EI)_{HG}}$$

w_{HG}: hull girder deflections

x: hull girder longitudinal axis

$(EI)_{HG}$: hull girder flexural rigidity

$q = D(x) - P(x) = Displacement - Weight$

Basic assumptions:

- no shear deformations
- no contribution from elastic foundation

Double Bottom Deformations

Bernoulli beam

$$\frac{d^4 w_{TF}}{dx^4} = \frac{\rho_w g T s_{TF}}{(EI)_{TF}}$$

w_{TF}: transverse floor deflections

x: transverse floor longitudinal axis

$(EI)_{TF}$: transverse floor flexural rigidity

ρ_w: water mass density

T: ship's draft

s_{TF}: transverse floor spacing

$W_{TF} \sim L^4{}_{TF} \sim [f(B)]^4$

L_{TF}: length of the floors

B: ship's beam

- Grillage consisting of transverse floors and longitudinal girders

- bending and shearing deformations retained

- clamped boundary conditions along

  - aft engine room transverse bulkhead
  - forward engine room transverse bulkhead

******

Calculations made for a TANKER

L = 300m, Beam = 50m, Depth = 25.5m

Loaded Draft = 19m. Deflections corresponding to a draft difference of 11.5m:

Bending Deflections (downwards)

<<

Shearing Deflections (upwards)

Transverse Web Frame Deformations

- frame analysis

- $w_{TWF} = w^1{}_{TWF}\left(\dfrac{\text{length of transverse floors in wav of gears}}{\text{length of webs of side shell}}\right)$

  $= w^{11}{}_{TWF}$

w_{TWF}: transverse web frame deflections

of an inch, the FEM is the only method of structural analysis
which can lead to sufficiently reliable and accurate results,
and for this reason it is included in the method being proposed
here. The method is described in detail in various textbooks,
e.g. [54,57] and only those aspects which are of particular
relevance in the present study will be briefly addressed in
this section.

The FEM has the advantage of enabling the structural
engineer to construct a model which can follow very closely the
exact geometry and material properties of the structure being
examined. The major limitation of the method is the large
computer cost it can imply, particularly if a large number of
elements is used and if a nonlinear material analysis is to
be performed. In addition to a large computer cost, the time
involved in input preparation is extremely large and this
represents another shortcoming. In the present case, the authors
believe a FEM analysis (using a fine mesh) should be used at
the last design stage when the structure is well defined, after
some parametric studies have been performed with the simpler
methods discussed previously. The FEM can in fact be used to
tune and verify the results obtained from simpler methods.

A difficulty in implementing the FEM is the choice of the
structural model, in particular the types of elements to be
used. An exact representation of a complex structure such as
a ship would involve so many elements that a certain degree
of simplification becomes imperative. In the present case,
the results we are looking for are deflections (translations
and rotations) within the support structure of the machinery
components, in particular at the critical points defined in
Fig. 20, in the case of geared steam-powered vessels. In
general, in studies of this nature the FEM model can represent
the ship structure from the after end to the forward bulkhead
of the machinery space. Due to symmetry, only the port or
starboard side have to be considered, except if the structural
arrangement is not symmetric about the ship's center plane.
An initial analysis with a coarse mesh representing the whole
machinery space, followed by a fine-mesh analysis of the struc-
ture in the area of interest, say in the vicinity of the reduc-
tion-gear casing, can be an efficient way of carrying the
analysis, as suggested for example in [58].

The element types to be used should ideally be thin-
shell elements. These are, however, expensive to implement so
that simpler models are usually adopted. A combination of
plane stress, truss and beam elements is often used. This
implies that some structural components have to be lumped to-
gether, which requires a good degree of judgement. This is
a subject which is not very deeply covered in the literature,
and in which opinions vary quite widely since no exact rules
can be established.

Another important decision which has to be made regarding the structural model is the definition of the boundary conditions. At the forward bulkhead of the machinery space, full fixity can, in general, be assumed. Along the center plane, the conditions of symmetry or anti-symmetry are used. If a coarse and a fine-mesh analysis are carried in sequence, the results from the coarse-mesh analysis are used to derive the most appropriate set of boundary conditions to be adopted at the fine-mesh stage.

The application of the FEM to the hull machinery compatibility problem is illustrated at the end of the chapter, when applying the proposed method to a specific ship.

4. Proposed Method

In order that the requirements for hull machinery rigidity compatibility are adequately met, the designer has to devote particular attention to the design of the shafting system and the structural arrangement of the machinery support systems.

In terms of the shafting system, the main criterion, as discussed earlier, is the maximum allowable difference between bearing reactions or ΔR. This is used to estimate the required number of bearings and their corresponding location. The bearing unit loads, the span to shaft diameter ratio, the shafting flexibility and the lateral vibration natural frequencies are additional factors to be considered in determining the number and location of shaft bearings. Using the Boston Naval Shipyard Code [56], described in Appendix B, a continuous-beam-shafting analysis can be made to determine the reactions, slope, deflection, bending moments, and the corresponding influence numbers. The definition of suitable alignment criteria in the vertical and horizontal plane is then used together with the results of the shafting calculations to define suitable alignment tolerances under the various operating conditions. The appliction of the ΔR criterion is summarized in the diagram in Figure 32.

The second main area of concern to ensure adequate hull-machinery compatibility is the structural design of the machinery support systems. The general guidelines given in Chapter II should be followed when designing the foundation. The deflections at the critical support points should then be determined by using sequentially the two methods discussed earlier. The stress-hierarchy method, in particular the frame and the grillage models, should be used to produce parametric variations involving the main structural parameters which define the structural arrangement (see Table II). In particular, the floor and girder scantlings and spacing and the frame and web frame scantlings and spacing should be varied. By doing so, the designer can determine the most efficient way of achieving the stiffness required by the machinery manufacturers. It is

APPLICATION OF THE ΔR CRITERION

$$R^o = R^A + I \Delta$$

$$\Delta = F P$$

$$R^o = R^A + I F P$$

$$0 < R^\cdot < R^P$$

$$-\Delta R < R^o_1 - R^o_2 < \Delta R$$
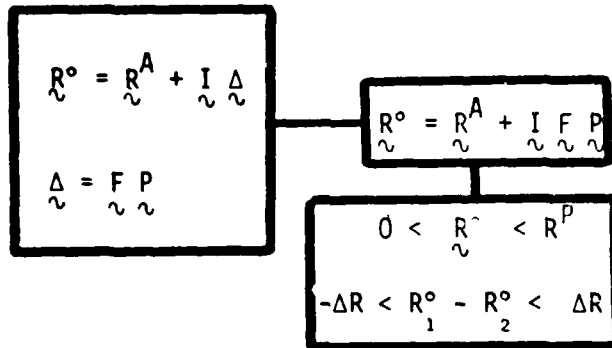
R^o : vector of bearing reactions during operation

R^A : vector of bearing reactions during alignment

I : matrix of influence coefficients

F : engine room double bottom flexibility matrix

Δ : engine room deflections at bearing locations

P : applied loads

R^P : limits of magnitude of bearing reactions based upon allowable bearing pressures

ΔR : maximum allowable difference between bearing reactions No. 1 and No. 2

$I = I$ (Bearing number and location, shaft rigidity)

$F = F$ (Hull (GIRDER) bending and shear rigidity,
Engine room transverse web (FRAME) rigidity,
Engine room double bottom (GRILLAGE) bending and
shear rigidity)

Figure 32

obviously necessary to tune the frame and grillage models so
that the results can be accepted with a certain degree of con-
fidence. This can be achieved by using the FEM for which the
degree of accuracy is much higher. After the designer is fairly
confident that the structural arrangement achieves the required
stiffness, then a final check with the FEM can also be under-
taken.

The deflections at the critical support points should
be computed for different loading conditions, in particular
the light loading condition and the full loading condition,
with full thrust and torque being applied to the thrust bearing
foundation. Symmetrical and antisymmetrical loads should
also be considered. All loads are applied statically.

The various components of the SHM have been inplemented
in the computer, as described in Appendix C. The software
package called ANALYSIS consists of the following two modules:

- GIRDER

- SPACE

The GIRDER module is for hull girder analysis. The SPACE
module is for two-dimensional frame analysis and for grillage
analysis of the double bottom. A listing of the computer code
analysis is given in Appendix C, together with a description
of the main variables of the program. It should be noted
that the SPACE module is in fact a three-dimensional frame
analysis program, so that the structure being analyzed does
not necessarily have to be a two-dimensional frame or a grillage.
A three-dimensional frame containing all the main parameters
given in Table II can in particular be considered, and this
certainly represents a very powerful model.

The method briefly described in the foregoing is
summarized in Table XI. It can best be understood by consider-
ing the specific example discussed in the next Chapter.

TABLE XI

Hull-Machinery Compatibility
Proposed Methodology Summary

Shaft Design	Foundation Design
● Meet ΔR criterion (see Figure 32)	● Follow whenever possible the general guidelines of Chapter II, Section 2.
	● Construct SHM models and tune using FEM so that computed deflections are comparable.
	● Perform parametric variations with SHM, varying the main parametric variations listed in Table II.
	● Choose combination of parameters which meets requirements given in Chapter III. Make this choice on the basis of a suitable efficiency criterion, such as material (weight) savings.
	● Check final design with FEM.

CHAPTER V.   EXAMPLE OF APPLICATION

1. Ship Main Characteristics

The ship to be considered here is a 188,500-DWT tanker
(#11 in Tables III through VIII of Chapter II). For conven-
ience, its main characteristics are summarized in Table XII.

The tanker's main compartments, as shown in Fig. 33
include a forepeak compartment, five cargo tanks, ballast
water tank, slop tanks, engine room, and aft peak compartment.
A double bottom extends through the vessel's length, and two
longitudinal bulkheads 39' off the center line also extend
through the length. The framing system is transverse.

As shown in Fig. 33, the engine room is located aft,
just forward of the aft peak, and it extends from frames 71
to 114, over a length of 107.5'. At the forward end it has
a pump room. Its structural arrangement includes transverse
plate floors 0.75" thick at every frame, 30" apart, in way
of turbine foundations, reduction gear, thrust bearing, and
aft until the stern tube. Forward of frame 88, transverse
plate floors 0.63" thick are provided at every frame.

In the engine room, web frames have a maximum spacing of
five frames (150"). The scantlings of a typical web frame
in way of the reduction gear are: web depth 4.25', web
thickness 0.63", flange width 6", flange thickness 1".
Transverse frame scnatlings in this same area are: web depth
10", web thickness 0.50", flange width 4", flange thickness
0.75". Side stringers are spaced 7.5' to 9', and typical
scantlings have: web depth 39", web thickness 0.50", flange
width 5", and flange thickness 0.50".

Flats are 30', 45', and 60' above the base line, extend-
ing for the complete length of the engine room (see Fig. 34).
There are four stanchions (12" x 12" x 1.59" WF) 15' off
center line, port and starboard at frames 88 and 99, extending
from the inner bottom to the 60' flat (see Fig. 34 and 35).
A sloping flat extending from the engine room aft bulkhead
to second reduction gear and close to the thrust bearing is
provided. The sloping flat is 3' wide on each side of center
line and broadens into the gear foundation at frame 96. A
bilge-water oil drain tank is situated below this sloping
flat and extends from frame 96 to frame 110.

The transverse watertight bulkhead forming the forward
end of the engine room, at frame 71, is stiffened by bulb
angles having the following scantlings: above the 45' flat,
16" x 6" x 0.50"/0.75", and below the 45' flat 24" x 8" x
0.50"/1". The plate thickness varies from 0.50" to 0.75".

TABLE XII

TANKER MAIN CHARACTERISTICS

Principal Dimensions

Length BP	915'
Breadth	166'
Depth	78'
Draft	59'

Displacement   188,500 DWT

Machinery

Steam turbine   28,000 SHP

Engine Room Construction

Transverse framing, spacing 30"

Engine room length 107.5'

Engine room width in way of
    reduction gear 41.7'

Web frame spacing 150"

Inner bottom plate 0.72" thick

Bottom C. L. girder 0.81" thick

Bottom side girders 0.63" thick

Double bottom depth 9'

Shafting Details

Line shaft diameter 23.75"

Tail shaft diameter 29.75"

Thrust bearing location 4' aft of
    reduction gear

Number of line shaft bearings, 1

Height of thrust bearing center
   above inner bottom 8.93'

TANKER'S MAIN COMPARTMENTS

PROFILE

UPPER DECK

TYP. TRANSV. BHD.

TYP. TRANSV. FRAME
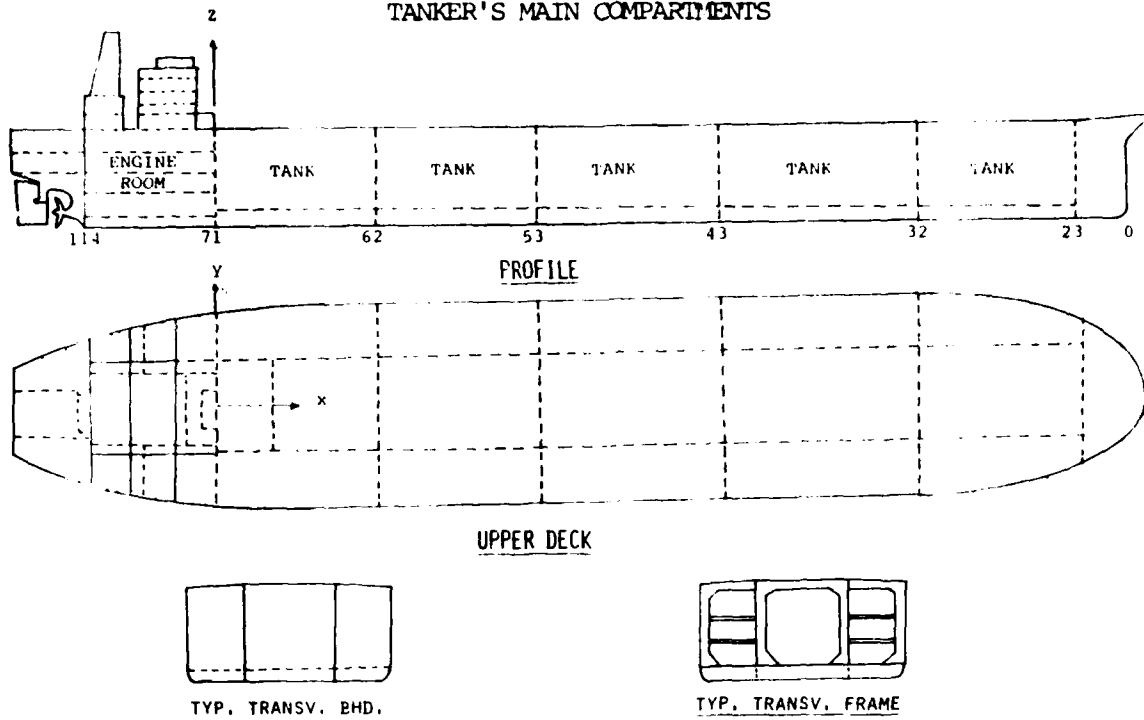
Figure 33

TANKER'S MACHINERY COMPARTMENT GEOMETRY

Figure 34

MAIN DECK

60' FLAT

40' FLAT

30' FLAT

STANCHION

15'

H.P. TURBINE FOUNDATION

STRGR.

INNER BOTTOM

L.P. TURBINE FOUNDATION

9' ABL

LONGITUDINAL GIRDERS
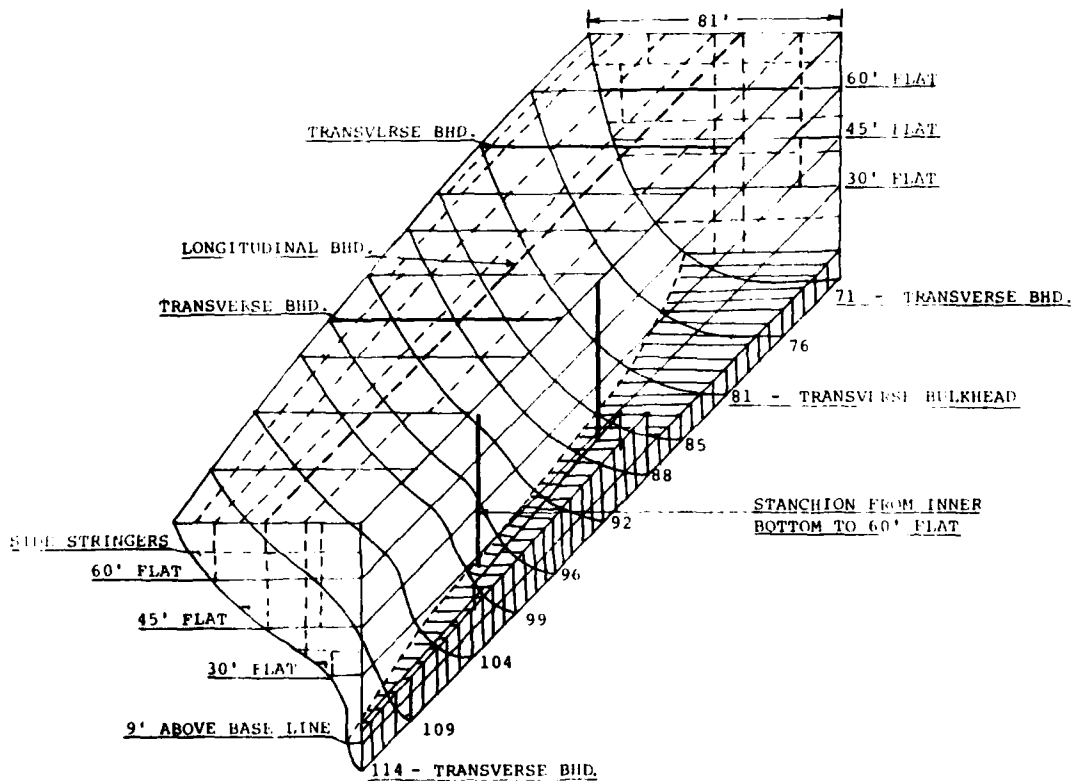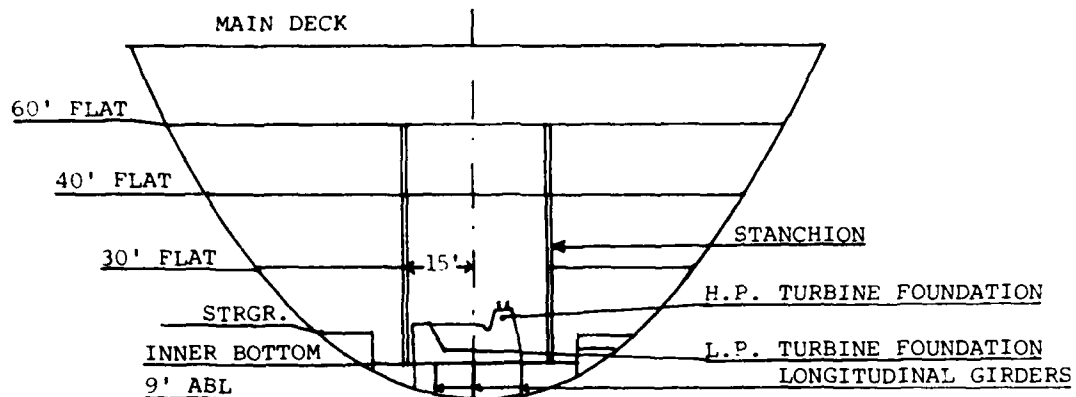
SECTION AT FR.88 - LOOKING FWD.

MAIN DECK

60' FLAT

45' FLAT (OPEN 15' OFF C.L.
P & S FR. 90-99)

30' FLAT

FLAT 3' WIDE EXTENDING FULL
LENGTH OF SHAFT

INNER BOTTOM

BILGE WATER OIL DRAIN TANK

9' ABL

LONGITUDINAL GIRDERS
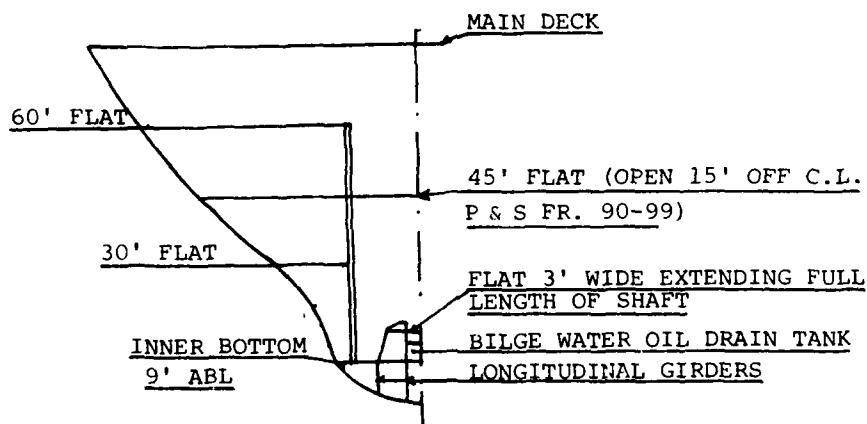
SECTION AT FR. 99 - LOOKING FWD.

(PORT SHOWN - STBD. SIMILAR)

Figure 35

The transverse bulkhead forming the aft end of the engine room, at frame 114, is stiffened by bulb angles having the following scantlings: 14" x 4" x 0.50"/0.75". The plate thickness varies from 0.44" to 1". The side shell plating thickness varies from 0.69" to 1.13", the main deck plating forward of frame 90 is 0.88" thick, and aft of this frame it is 0.72" thick.

In way of the reduction gear, there are a total of five longitudinal bottom girders, at center line and 9' and 18' off center line on port and starboard. The girder plate thickness is 0.81" for the center girder and 0.63" for the side girders. The inner bottom plating is 0.72".

2. Finite-Element Model

A FEM for the hull structure aft of the forward bulkhead of the machinery space was developed for the purpose of obtaining accurate results for the deflections at the critical support points defined in Fig. 20, in particular at the corners of the reduction gear foundation. The program ADINA (Automatic Dynamic Incremental Nonlinear Analysis) was used for implementing the FEM analysis [59]. A special graphics package to display the input mesh and to check the grid geometry was developed by the authors as part of this research effort.

The complete hull structure from frame 71 to the transom was included in the model. Vertically, the model extends from the ship's bottom to the main deck and in the transverse direction from the center plane to the port side shell plating. Only the portside half of the afterbody was modelled, since the structure can be for any practical purposes considered symmetric. The superstructure was not included, since it does not affect to a large extent the deflections in the region of interest. Other structural elements having a negligible influence on the double-bottom deflections were also omitted. This was necessary in order to keep the computer time required for the analysis within reasonable limits.

The FEM contains a total of 765 elements subdivided among the following groups: 6 truss elements, 405 plane stress three-dimensional orthotropic elements, and 354 beam elements.

The truss elements were used to model the stanchions. The plane-stress elements were used to model the transverse and longitudinal bulkheads, decks, and flats. The linear-orthotropic-elastic model was used in order to represent in a simple way the effect of the stiffeners, and by using this approach it was possible to decrease substantially the number of elements needed to represent adequately the structure. The

computation of the equivalent orthotropic material constants was done in accordance with the classical orthotropic-plate theory [60].

The beam elements were used to model the double-bottom structure and side shell, including transverse web frames, bottom plating, inner bottom plating, transverse floors, and longitudinal girders. The side shell plating was accounted for in the stiffness of the side stringers, by considering an effective breadth of plating attached to a beam element.

The finite-element grid includes a total of 513 nodes. The mesh is shown in Fig. 36, as plotted by the graphics package developed by the authors. Fig. 37 shows a plot of the longitudinal bulkheads and Fig. 38 a plot of the flats.

The following boundary conditions were adopted: full fixity at the forward engine room bulkhead (frame 71) and symmetry conditions along the center line (zero y displacement and zero rotations about the x and z axes, the coordinate system being the one shown in Fig. 33). The total number of degrees of freedom is 1806.

Two static-loading conditions were considered: full-load condition at full power (full thrust and torque), and light-load condition. The hydrostatic pressures, machinery component weights, fuel weight, etc., were translated into statically equivalent forces and moments to be applied at the grid's nodal points. ADINA has provisions for gravity loading due to the mass of the elements, and the material specific gravity was adjusted so as to provide the total correct structural weight.

3. Results

3.1 Shaft Bearing Reactions

The shafting system of the tanker under consideration has been studied using the Boston Naval Shipyard Computer Code [56]. The shafting arrangement is shown in Figure 39. The bearing reaction influence numbers, as well as the straight line bearing reactions have been obtained for various magnitudes of the shaft length L and diameter D. The corresponding numerical values for the particular case of $D = 23.75"$ and $L/D = 15$ are presented in Table XIII. This information is useful in order to examine the influence of the movements of the bearing locations on the changes of the magnitude of the bearing reactions. In the alignment condition, the equality of the magnitude of the reactions at bearings No. 1 and 2 is sought.

TANKER FINITE-ELEMENT GRID
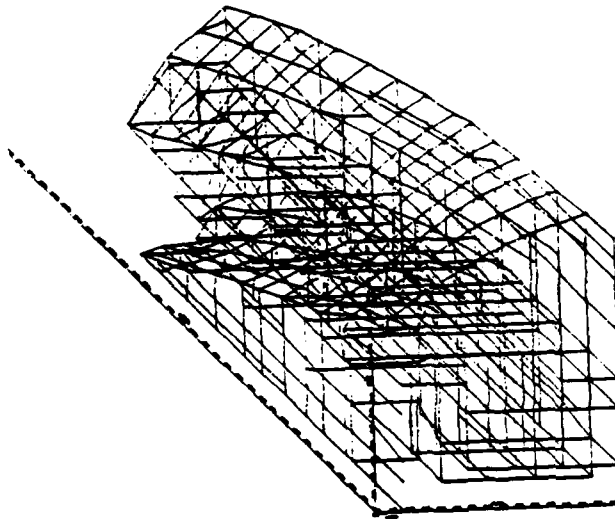

Figure   36

FINITE-ELEMENT GRID ~ FLATS
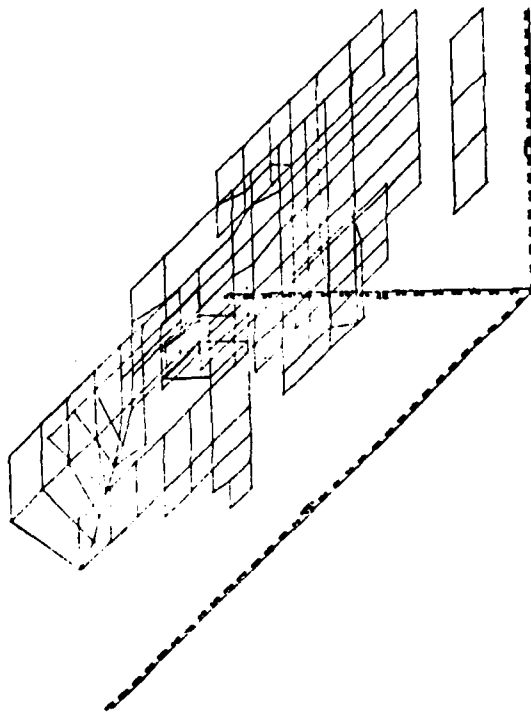


Figure   37

FINITE-ELEMENT GRID - LONGITUDINAL BULKHEADS
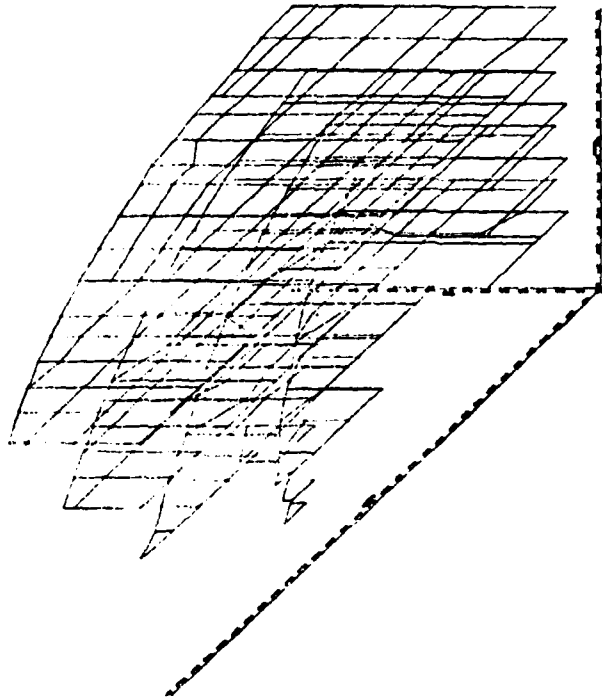
Figure 38

SHAFTING ARRANGEMENT


107,400 lbs.

178,100 lbs.

L/D = 15

908.17
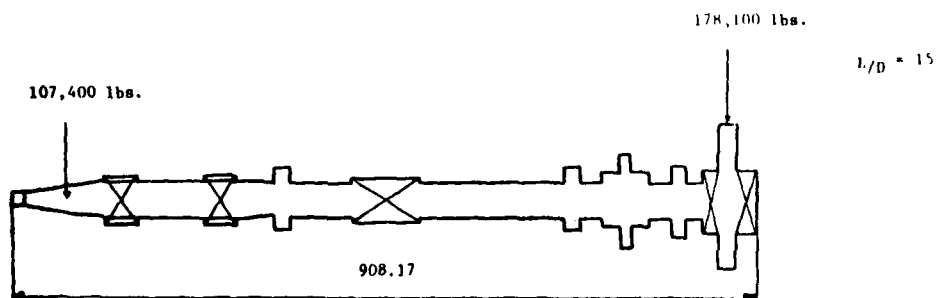
Figure 39

TABLE XIII

TANKER L/D=15.00 D=23.75

Bearing Reaction Influence Numbers

(lb. per 0.001 inch rise of bearings)

Bearing No.	1	2	3	4	5
1	1524.2	-1942.9	718.0	-397.5	98.2
2	-1942.9	2500.2	-1010.8	602.3	-148.8
3	718.0	-1010.8	1068.6	-1274.5	498.8
4	-397.5	602.3	-1274.5	1919.3	-849.6
5	98.2	-148.8	498.8	-849.6	401.5

Straight Line Bearing Reactions

(lbs.)

45625.2	174149.4	54682.9	-50552.5	207918.5

Tables XIV, XV and XVI list the values of the straight line bearing reactions R_1 and R_2, the influence coefficients I_{11} and I_{22} and the magnitude of the ASE (Allowable Setting Error) based on an assumed ΔR equal to 15,000 lbs for various values of the L/D ratio. The bearing reactions are given in lbs and the influence coefficients in lbs per unit of an inch. The ASE is computed in mils of an inch. To avoid incompatibility problems the value of the actual $(R_1 - R_2)/(I_{11} - I_{22})$ must be kept smaller than the corresponding value of ASE for a given allowable ΔR.

3.2 Hull Girder

The four loading conditions (for the tanker under consideration) presented in Table XVII have been analyzed. They are:

- the lightship

- the lightship plus segregated ballast

- Martinez departure (ballast)

- Alaska departure (full load)

Table XVII also lists the cargo carried, the mean draft, the maximum bending moment and the bending deflections amidships for each of the four loading conditions listed above. The hull girder deflections are considered positive when upwards from that reference line. The hull girder deflections amidships obtained from the proposed procedure are compared with the predictions based on the semiempirical method suggested in Reference [12]. It can be seen from Table XVII that the two results compare fairly well.

The hull girder deflections are also plotted in Figure 40 within the extent of the machinery compartment from frame 71 to frame 114. It can be seen from Figure 40 that the lightship, the lightship plus segregated ballast and the Martinez departure are in a hogging condition, whereas the Alaska departure is in a sagging condition. The hull girder deflections plotted in Figure 40 are total deflections (bending plus shear deflections).

TABLE XIV

ALLOWABLE SETTING ERROR (D-23.75 in.)

L/D	R_1	R_2	I_{11}	I_{22}	ASE
10	67425.4	138000.9	2339.7	4468.5	- 7.046
12	58848.7	153234.1	1936.6	3435.7	-10.006
13	54455.7	160429.8	1777.9	3062.5	-11.677
14	50044.9	167387.3	1641.6	2755.4	-13.467
15	45625.2	174149.4	1524.2	2500.2	-15.369
16	41205.2	180744.8	1422.9	2287.0	-17.359
16.16	40501.6	181781.9	1408.1	2256.4	-17.682
17	36817.8	187154.6	1336.4	2109.9	-19.392
18	32619.4	193180.0	1265.1	1967.6	-21.352

R_1, R_2 straight line bearing reactions in LBS
I_{11}, I_{22} LB. per 0.001 rise of bearings

TABLE XV

ALLOWABLE SETTING ERROR (D-22.5625 in.)

L/D	R_1	R_2	I_{11}	I_{22}	ASE
10	69333.4	134276.6	2423.3	4721.0	- 6.528
12	61352.9	148742.1	1987.2	3594.5	- 9.332
13	57222.3	155586.3	1814.3	3185.8	-10.937
14	53082.0	162175.8	1665.8	2849.2	-12.675
15	48948.5	168547.6	1537.6	2569.2	-14.541
16	44823.8	174743.2	1426.7	2334.6	-16.522
17	40703.0	180797.7	1331.0	2137.6	-19.597
18	36617.0	186690.6	1249.5	1974.0	-20.704
19	32741.4	192192.2	1183.6	1844.4	-22.700

R_1, R_2 straight line bearing reactions in LBS
I_{11}, I_{22} LB. per 0.001 rise of bearings

TABLE XVI

ALLOWABLE SETTING ERROR (D-21.375 in.)

L/D	R_1	R_2	I_{11}	I_{22}	ASE
10	71022.5	130887.6	2520.8	5013.1	- 6.019
11	67455.7	137817.3	2266.9	4330.5	- 7.269
12	63668.5	144570.1	2047.4	3779.1	- 8.662
13	59789.3	151088.7	1858.4	3329.3	-10.198
14	55897.8	157348.6	1695.8	2958.8	-11.876
15	52030.3	163363.9	1555.6	2650.7	-13.697
16	48197.1	169166.9	1434.2	2392.3	-15.656
17	44380.6	174815.1	1328.9	2174.3	-17.743
18	40568.8	180348.9	1237.9	1990.4	-19.934
19	36770.9	185771.1	1160.5	1837.4	-22.160
20	33160.6	190852.1	1098.7	1717.4	-24.244

$R_1 R_2$ straight line bearing reactions in LBS

I_{11}, I_{22} LB. per 0.001 rise of bearings

HULL GIRDER DEFLECTIONS
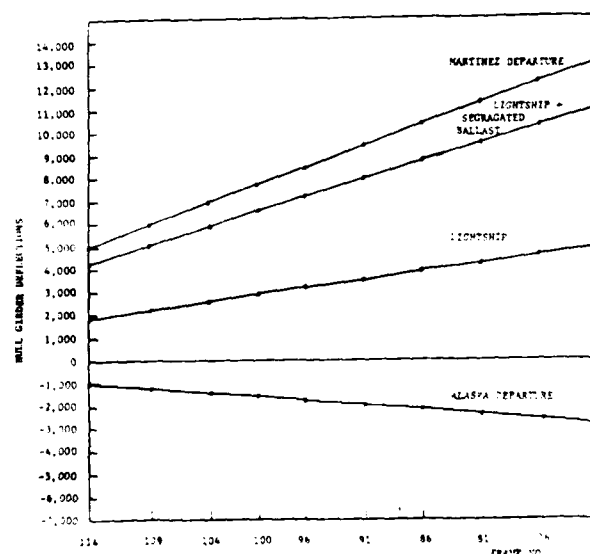
Figure 40

TABLE XVII

HULL GIRDER DEFLECTIONS

Loading Condition	Cargo Tons	Mean Draft in Ft	Maximum Bending Moment FT-TONS	Bending Deflection Amidships* MILS of IN	Bending Deflection Amidships** MILS of IN
Lightship	-------	9.06	-790,346	8,005	9,097
Lightship + Segregated Ballast	To meet I.M.C.O. Require- ments	25.64	-1,705,970	17,278	19,241
Martinez Departure	59,857 S.W.	26.87	-2,077,382	21,040	23,115
Alaska Daparture	182,346 Oil	59.01	657,075	-6,655	-5,450

* estimated from REF [12].

** from proposed SHM.

3.3 Transverse Frames

The deflections of the transverse frames 92, 96 and 109 have been obtained due to weight and hydrostatic pressure loading. The geometry of the frame 92, 96 and 109 is depicted in Figure 41, 42 and 43 respectively. The centerline nodal points are not allowed to displace horizontally for symmetric loading conditions. Further-more, the rotations at the centerline nodal points are zero for symmetric loading. The deflections of the lowest center-line nodal point are presented in Table XVIII for three waterlines with mean draft equal to 59.33, 65.27 and 53.40 feet. The deflections reported in Table XVIII are with respect to the deflections of the highest centerline nodal point. Deflections are given in mils of an inch and are considered positive upwards. The transverse web frame structure is compressed by the hydrostatic pressure from the bottom and as a result the double-bottom structure

108
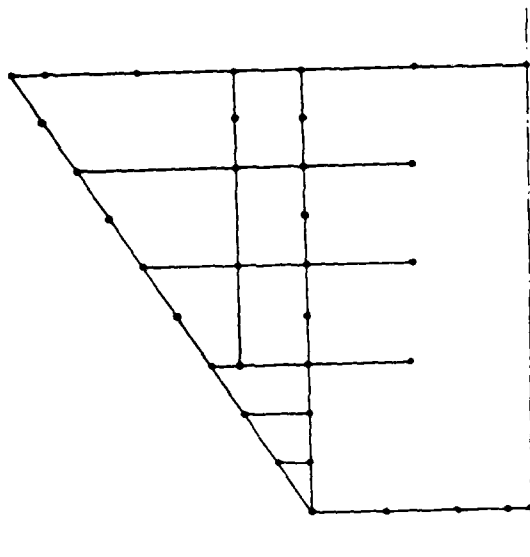
TRANSVERSE FRAME 92
GEOMETRY

Figure 41

TRANSVERSE FRAME 96
GEOMETRY

Figure 42
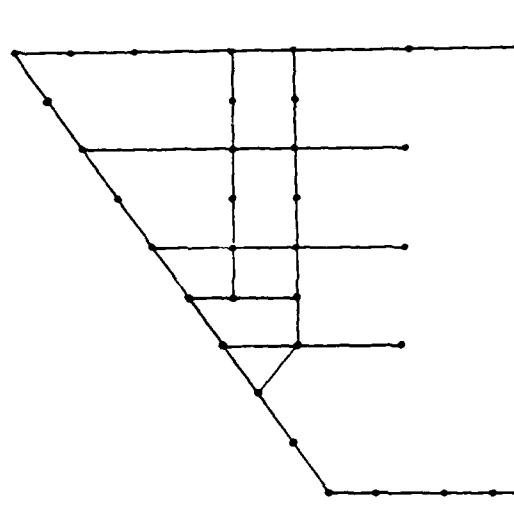
TRANSVERSE FRAME 109
GEOMETRY

Figure 43
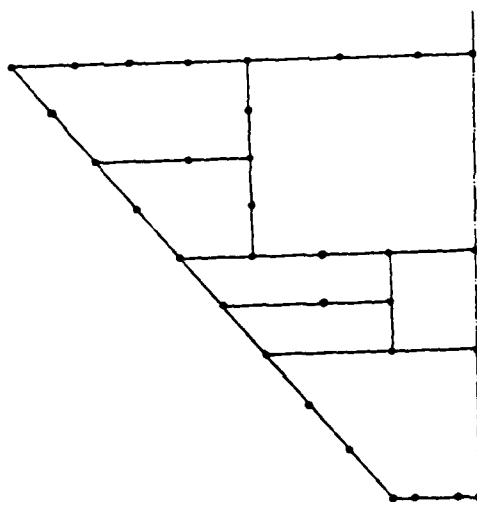
has a tendency to displace upward. However, the hydro-
static pressure at the side plating tends to produce
downward displacements at the bottom structure. It can
be seen from Table XVIII that for the frames 92 and 96 the
centerline displacements calculated are directed downwards
indicating that the side plating pressure deformation
mechanism is the dominant one, in this case. The center-
line deflection results for frame 109 suggest that, in that
case, the hydrostatic bottom pressure deformation mechanism
is dominant.

TABLE XVIII

BEARING REACTION INFLUENCE NUMBERS

Frame No. / Mean Draft ft	59.33	65.27	53.40
92	-20,230*	-26,033	-13,152
96	- 7,739	-11,894	- 2,873
109	5,654	5,716	6,015

* deflections given in MILS of an INCH and considered
positive upwards

Deflections for a sample case have been computed to
consider the case of asymmetric pressure distribution (the
theoretical procedure is presented in section 2.2.2). The
port and starboard water levels considered are d_1 = 53.40'
and d_2 = 65.27' respectively (Fig. 31) with a mean draft
equal to 59.33'. The centerline vertical deflection for
frame 96, under the loading condition mentioned above, has
been found to be equal to 4,970 mils of an inch.

3.4 Double Bottom

The engine-room double-bottom structure, for the tanker
under consideration, has been modelled as a grillage consisting
of interesecting beams (transverse floors and longitudinals).
Fig. 44 presents the geometry of the grillage structure
analyzed by the SHM. The deflections of the grillage structure
have been obtained due to weight and hydrostatic loading. The
grillage centerline vertical deflections are plotted in Fig. 45
for three different draft levels. The deflections are considered

GRILLAGE CENTERLINE VERTICAL DEFLECTIONS

T DRAFT

T=65.77'

T=59.33'

T=53.40'

MILS
OF INCH

300

250

200

150

100

50

GRILLAGE CENTERLINE VERTICAL DEFLECTIONS

71 76 81 80 88 86 91 96 100 104 109 114

FRAME NO.

Figure 45

ENGINE ROOM DOUBLE-BOTTOM GRILLAGE STRUCTURE

FRAME
NUMBER

71

76

80
81

85

88

92

96

99

104

109

144
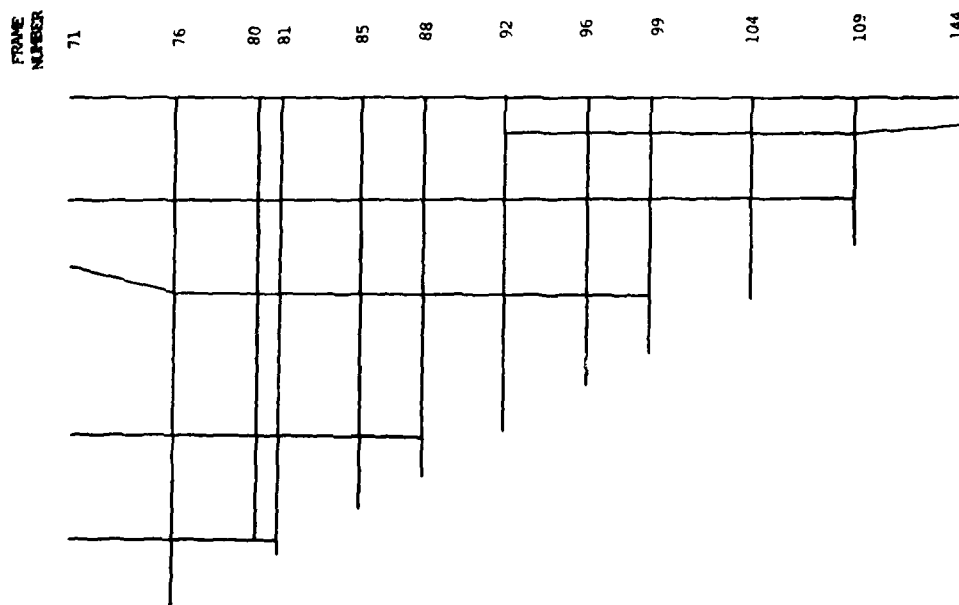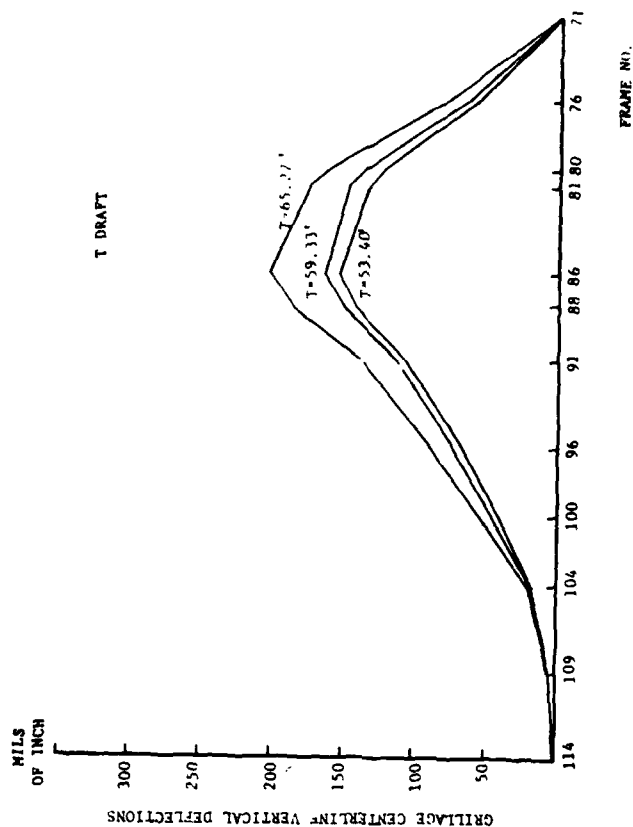
Figure 44

positive when upwards. The magnitude of engine-room double-bottom centerline deflections can be obtained from Fig. 45 as well as its dependence on waterline draft variations.

3.5 Finite-Element Model Results

The vertical deflections along the ship's centerline as determined by the finite-element analysis are shown in Fig. 46. These deflections are measured relatively to the forward bulkhead of the machinery space, since in the analysis a perfect clamped condition was assumed along this bulkhead. It can be concluded from Fig. 46 that in the light condition the hull structure along the machinery space deforms in hogging. In the full load condition, the forward part of the machinery space deforms in sagging (from frames 71 to 76), while the after part (aft of frame 76) deforms in hogging. This inversion of curvature within the machinery space in the full load condition was also observed in the studies reported in [2], as mentioned in Chapter I.

The most relevant result the finite-element analysis provided concerns the deformations at the critical points defined in Fig. 20, in particular at the corners of the reduction gear foundation. The relative deflections at these points between the full and light-load conditions are shown in Fig. 47. The results indicated in this figure show that due to the increase in draft and the applied thrust and torque, the four critical points (forward and aft port corners and low-speed gear bearings), suffer a translation forward, along the x-axis, which does not depart significantly from an average of 0.037".

The translation along the z-axis is larger for the forward low-speed gear bearing (0.0699"), and much smaller for the after corner (0.0341"), while for the remaining two points the values are closer (0.0466" and 0.0632"). Due to the condition of symmetry, the points along the center line do not move in the y direction, while the outer corners move by practically the same amount (0.0057" and 0.0058").

As discussed in Chapter III, the proposed deflection limits for foundations of geared-turbine-propulsion units, specify a maximum allowable value of 0.004" for the vertical movement at the seating surface of the forward port corner of the reduction-gear foundation relative to a plane passing through the remaining three critical points (the aft port corner, the forward low-speed gear bearing and the aft low-speed gear bearing). The finite-element analysis provides for each loading condition the distorted positions for the four points mentioned above. If the x, y and z coordinates for the forward and aft port corners of the reduction gear foundation and the forward and aft low speed gear bearings are denoted by x_i, y_i, z_i with $i = 1, 2, 3, 4$ respectively, then it can easily be shown that the vertical distance mentioned above can be obtained

VERTICAL DEFLECTIONS ALONG CENTERLINE (MILS)
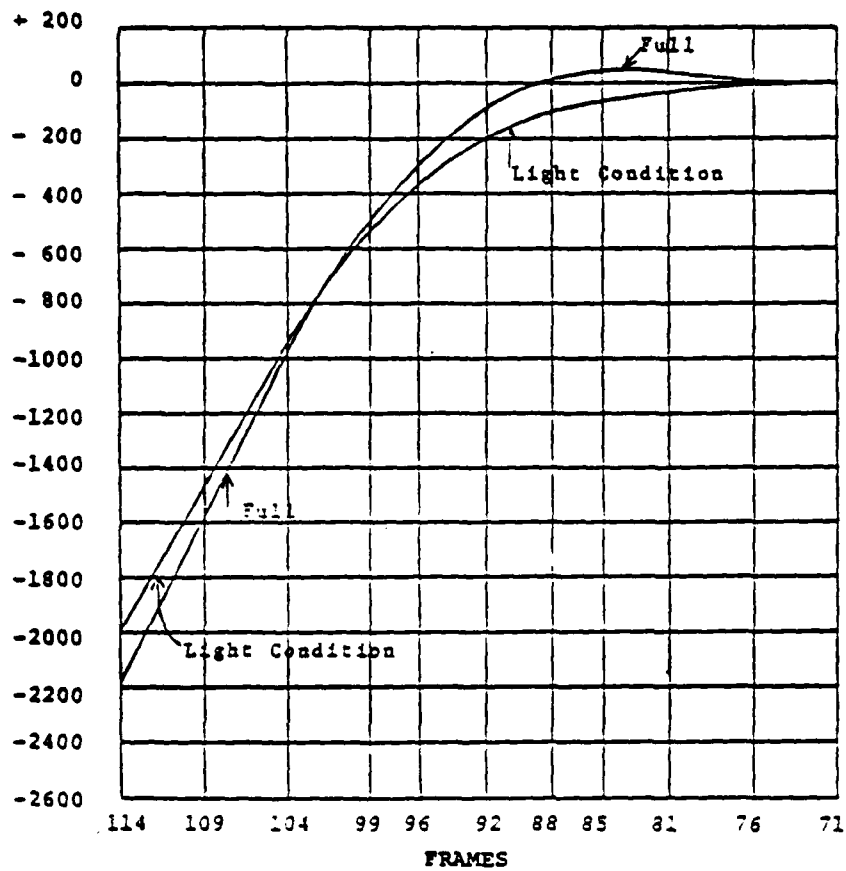
Figure 46

RELATIVE DEFLECTION DUE TO CHANGE IN DRAFT (FULL-LIGHT)
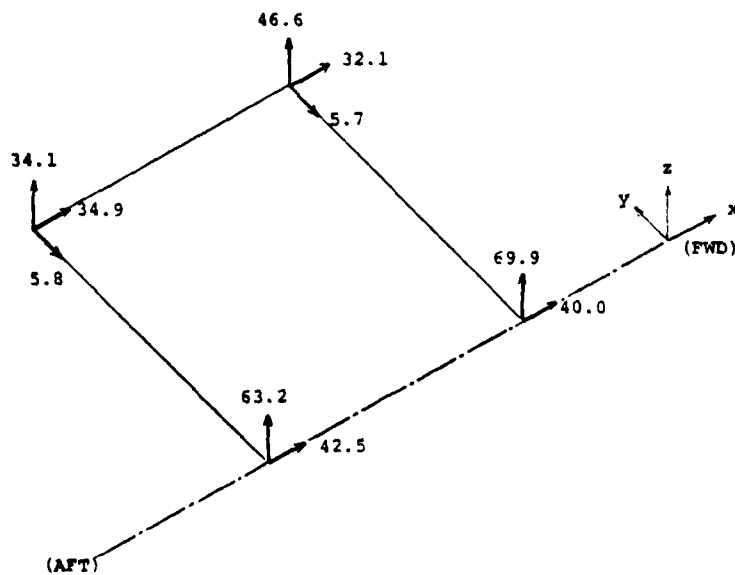

Figure 47

from the following equation:

$$d = \frac{Ax_1 + By_1 + Cz_1 + D}{\sqrt{A^2 + B^2 + C^2}}$$

where

$$A = (y_3 - y_2)(z_4 - z_2) - (y_4 - y_2)(z_3 - z_2)$$

$$B = (z_3 - z_2)(x_4 - x_2) - (x_3 - x_2)(z_4 - z_2)$$

$$C = (x_3 - x_2)(v_4 - y_2) - (v_3 - y_2)(x_4 - x_2)$$

$$D = -x_1 A - y_1 B - a_1 C$$

The value of d as determined from the equation defined above was computed for the two loading conditions considered here. It was found that in the light condition d is equal to 0.018", while in the full-load condition it is equal to 0.017". Thus, the difference between the two loading conditions gives a difference of 0.001" in the out-of-plane deformation suffered by the reduction-gear seating surface, which means that the design is well within the maximum limit of 0.004" suggested in Chapter III.

VI. CONCLUSIONS AND RECOMMENDATIONS

The potential for incompatibility of hull and foundation deflections at the support points of gear-turbine and diesel machinery relative to the rigidity requirements for the machinery should be given careful attention during the design phase, particularly in large vessels.

The problem is strongly influenced by the hull shape and beam in way of the machinery space. A machinery space located aft with narrow beam, "U" or "V" sections and waterlines which converge is inherently a relatively stiff arrangement, particularly in the vertical direction. In contrast, hull shapes similar to those of cruisers and destroyers, with gradually rising buttocks and strut-supported propeller bearings, place the machinery further forward, generally on a relatively wide, flat bottom. Such vessels appear to have a greater risk of hull-machinery incompatibility.

Conformance with the rules of the Classification Societies per se does not guarantee freedom from the problem of hull-machinery incompatibility and each case requires specific analysis of the design.

Location of the machinery space and arrangement of the machinery foundations are extremely important. The integration of machinery foundations with basic hull structure must be carefully planned. Bulkheads, both transverse and longitudinal, as well as decks and shaft alleys may be utilized to improve overall stiffness and particularly to distribute thrust forces to the hull over as wide an area as possible.

Analytical methods are available to determine the estimated maximum deflections at critical machinery support points. These deflections represent movements between the condition when machinery is aligned (usually lightship, zero power output) and some other operating condition (usually maximum draft, maximum power) that produces the greatest deflections.

Proposed limits for the deflections of machinery support points have been included in this report and may be used for comparison with estimated structural deflections. Whenever possible, all potential machinery suppliers should be consulted regarding their individual requirements.

In the event that deflection limits are exceeded, consideration should be given to increasing the structural stiffness as necessary. Should this become impractical, or uneconomical, then steps may be taken to render the machinery less susceptible to foundation movements. This may involve trade-offs between structure and machinery costs. Although it is impractical to

construct gear cases that can **resist foundation movements,**
special designs, chocking **arrangements, and other devices may**
minimize the effects. Similarly the box girder type of
construction for large diesels adds significantly to the
vertical stiffness in way of the engine.

It will be of interest that some further work is
performed in the future to include the following important
aspects of the hull-machinery coupled response:

- nonsymmetric response using the finite-element method

- more work on diesel-engine-propulsion support systems

- dynamic response

- include plane-stress analysis subroutine in stress
  hierarchy computer code to treat deflections of
  transverse bulkheads

- thermal effects

- perform parametric variations

VII. REFERENCES

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ACKNOWLEDGEMENTS

APPENDIX A: REVIEW OF CLASSIFICATION SOCIETY RULES

The rules of the following Classification Societies are included in this study:

(a) American Bureau of Shipping [34]
(b) Lloyd's Register of Shipping [35]
(c) Det norske Veritas [36]
(d) Bureau Veritas [37]
(e) Germanischer Lloyd [38]

The review presented here does not obviously include an exhaustive discussion of the various rules. It deals strictly with requirements concerning the ship structure in way of machinery spaces, and only those aspects considered to be of particular interest to the specific problem of hull-machinery compatibility are included, sometimes in condensed form. A direct attempt to compare the rules will not be made, since it is well known that the philosophy of design on which each Classification Society bases its requirements is different and has its own merits.

Only double-bottom construction will be discussed, since this is the type of structural arrangement used in large ships. The rules usually give requirements for high-strength materials, but these will not be considered here, since these materials are not widely used in the machinery spaces of large commercial vessels. The structural arrangement of shaft tunnels and machinery casings, a subject covered in all the rules, will not be discussed since it is not particularly relevant for this study. The same happens with respect to watertight bulkheads, which are required by all the rules to form the forward and after boundaries of the machinery spaces, and the structural strengthening of openings in way of machinery spaces. Similarly, structural details such as those involved in the attachment of web frames to the inner bottom or the machinery bolting arrangements will not be considered, since they do not affect to a large extent the overall structural rigidity of the foundation. Besides, as stated earlier, the whole area of ship structural details has already been the subject of extensive research [31, 32], so that there is no need to review it again here.

Whenever possible, the main sections in which specific requirements are defined are identified in parenthesis, so that if desired the reader can refer to them for complete information.

A.1 American Bureau of Shipping

In view of the effect upon the structure of the necessary openings in the machinery space, the difficulty of securing adequate support for the decks, of maintaining the stiffness of sides and bottom and of distributing the weight of the machinery, special attention is directed to the need for arranging, in the early stages of design, for the provision of plated through beams and such casing and pillar supports as are required to secure structural efficiency; careful attention to these features in design and construction is to be regarded as of the utmost importance. All parts of the machinery, shafting, etc., are to be efficiently supported and the adjacent structure is to be adequately stiffened. In twin-screw vessels and in other vessels of high power, it will be necessary to make additions to the strength of the structure and the area of attachments, which are proportional to the weight, power and proportions of the machinery, more especially where the engines are relatively high in proportion to the width of the bed plate. A determination is to be made to assure that the foundations for main propulsion units, reduction gears, shaft and thrust bearings, and the structures supporting those foundations are adequate to maintain required alignment and rigidity under all anticipated conditions of loading (19.1)

The engines are to be seated directly upon thick inner bottom plating or upon thick seat plates on top of heavy foundations arranged to distribute the weight effectively. Additional intercostal girders are to be fitted within the double bottom to ensure the satisfactory distribution of the weight and the rigidity of the structure (19.3.2).

Boilers are to be supported by deep saddle-type floors or by transverse or fore and aft girders arranged to distribute the weight effectively. If the boilers are supported by transverse saddles or girders the floors in way of boilers are to be suitably increased in thickness and specially stiffened. Proper accessibility and ventilation have to be ensured, and the thickness of adjacent material is to be increased as required, where the clear space is less than recommended (19.5).

Thrust blocks are to be bolted to efficient foundations extending well beyond the thrust blocks and

arranged to distribute the loads effectively into the adjacent structure. Extra intercostal girders, effectively attached, are to be fitted in way of the foundations as may be required (19.7).

Shaft stools and auxiliary foundations are to be of ample strength and stiffness in proportion to the supported weight (19.9).

Special provisions are given in the rules regarding the arrangement and scantlings of bottom structure in way of machinery spaces.

Solid floors are to be fitted on every frame under machinery and transverse boiler bearers. Their minimum thickness t is

$$t = 0.036L + 6.2 \text{ mm} \tag{1}$$

where L, the rule length is m in such that $L < 427$ m. Where boilers are mounted on the tank top the thickness of the floors and intercostals in way of the boilers is to be increased 1.5 mm above engine space requirements (7.3.4).

The inner bottom plating minimum thickness t in way of engine spaces is given by

$$t = 0.037L + 0.009s + 1.5 \text{ mm} \tag{2}$$

where the rule length is $L \leq 427$ m and s is the frame spacing in mm (7.5.1).

Under boilers, there is to be a clear space of at least 457 mm. Where the clear space is necessarily less, the thickness of the plating is to be increased as may be required (7.5.3).

In way of engine-bed plates or thrust blocks which are bolted directly to the inner bottom, the plating is to be at least 19 mm thick. The thickness is to be increased according to the size and power of the engine (7.5.4).

Side girders with minimum thickness t given by equation (1) are to be so arranged that the distance from the center girder to the first girder, between the girders or from the outboard girder to the center of the margin plate does not exceed 4.57 m. Additional full or half-depth girders are to be fitted beneath the inner bottom as required in way of machinery and thrust seatings and beneath wide-space pillars. Where the bottom and inner bottom are longitudinally framed, this requirement may be modified (7.9).

The rules stress the need to provide sufficient transverse strength and stiffness in the machinery space by means of webs, plated through beams, and heavy pillars in way of deck openings and casings (8.15).

'Tween-deck webs are to be fitted below the bulkhead deck over the hold webs as may be required to provide continuity of transverse strength above the main webs in the holds and machinery space (9.7).

Special support provided by stanchions or pillars or by means which are not less effective is to be arranged at the ends and corners of deckhouses, in machinery spaces, at ends of partial superstructures and under heavy concentrated weights (11.1).

Under boilers, the plating of effective deck is to be at least 15 mm in thickness (16.5.8).

For tankers, machinery spaces aft are to be specially stiffened transversely. Longitudinal material at the break is also to be specially considered to reduce concentrated stress in this region. Longitudinal wing bulkheads are to be incorporated with the machinery casings or with substantial accomodation bulkheads in the 'tween decks and within the poop (22.15).

A.2 Lloyd's Register of Shipping

Except where otherwise noted the rule sections given in parenthesis are taken from Part 3, Chapter 7.

The rules make a distinction between three basic locations for machinery spaces, namely: midship region, aft region with a cargo compartment between it and the after peak bulkhead, and aft region with the after peak bulkhead forming the aft end of the machinery space (1.1.2).

If the machinery spaces are amidships and the shell and decks outside the line of openings are longitudinally framed in way of adjacent cargo spaces, the machinery space is also to be longitudinally framed (1.2.2).

In longitudinally framed machinery spaces the maximum spacing S_{max} of transverses is given by

$$S_{max} = 3.8 \text{ m} \qquad\qquad L \leq 100 \text{ m}$$

$$S_{max} = (0.006L + 3.2)\text{m} \qquad L \geq 100 \text{ m}$$

where L is the rule length in m. In way of a machinery space situated adjacent to the after peak, the spacing is not to exceed five transverse frame spacings (1.2.4).

The rules emphasize the need for suitable structural continuity of the machinery spaces, suitable deck strengthening in way of machinery openings, and suitable support systems for deck beams (in transversely framed ships), deck longitudinals (in longitudinally framed ships) and machinery casings. Also, in way of concentrated loads such as those from boiler bearers or heavy auxiliary machinery, the scantlings of lower decks or flats must be specially considered taking the actual loading into account (1.3, 2.1, 2.2 and 2.3).

If the machinery space is amidships, web frames should be fitted and spaced not more than five frame spaces apart, and extending from the tank top to the level of the lowest deck above the load waterline. The scantlings should be such that the combined section modulus of the web frame and the main or 'tween deck frames is 50% greater than that required for normal transverse framing. These web frames can be omitted if the section modulus of the ordinary main or 'tween deck frames is to be increased by 50%, up to the level of the lowest deck above the load

waterline. Where fully effective stringers supported by web frames are fitted, the stringers may be considered as decks for the purpose of calculating the modulus of the frames (3.1.1 and 3.2.3).

Except where the machinery is adjacent to the after peak, longitudinal framing should have the same scantlings as for cargo spaces. For machinery spaces adjacent to the after peak, the section modulus Z of side longitudinals is given by

$$Z = 0.0065 \ ks\ell_e^2 \quad (h+0.167D) \quad cm^3$$

where k is higher tensile steel factor (equal to unity for mild steel), s is the spacing of floors and longitudinals in mm, ℓ_e is the effective length of the stiffening member in m, h is the load head in m, from mid-span to upper deck at side amidships (not less than 0.9m) and D is the rule depth (not more than 20 m) (3.1.2).

If the space is situated in the aft region and transverse framing is adopted, web frames are to be fitted in general not more than five frame spaces apart, extending from the tank top to the upper deck. A spacing up to seven transverse frame spaces is also acceptable if the ordinary frames are substantially strengthened to satisfy the overall modulus and inertia requirements. The scantlings of web frames below lowest deck and not supporting effective stringers are to be governed by the following minimum section modulus:

$$Z = 5KSh\ell_e^2$$

Above the lowest deck Z is given by:

$$Z = 1.68 \ CkTS\ell_e \quad \sqrt{D}$$

where S is the spacing or mean spacing of web frames
or transverse in m, C is a parameter taking the value
2.2 for a lower 'tween deck and 2.0 for an upper 'tween
deck, T is the rule draft and the remaining symbols have
already been defined. The minimum web depth to be used
in conjunction with these two expressions for Z is 2.5
times the depth of adjacent main frames (3.2.1 and Table
7.3.1).

If the span of ordinary frames below the lowest
deck or flat exceeds 6.5 m, one or more fully effective
side stringers are to be fitted to support the frames.
The scantlings are to satisfy the following minimum
section modulus Z:

$$Z = 7.75 \text{ kSH} \ell_e^2$$

where all the symbols have already been defined. The
minimum web depth in this case is two and a half times
the depth of adjacent main frames (3.2.1 and Table 7.3.1).
An arrangement of light stringers spaced about 2.5 m apart
may also be accepted as an alternative to the fully
effective stringers just defined (3.2.2).

If the machinery space is not in the after end
region, the web frames below the lowest deck supporting
effective stringers are to be found from the following
assumptions: fixed ends, point loadings, head to upper
deck at side, bending stress 93.2 N/mm^2 and shear stress
83.4 N/mm^2. Again the minimum web depth is equal to 2.5
times the depth of adjacent main frames (Table 7.3.1).

If the machinery is longitudinally framed, side
transverses are to be fitted. Below the lowest deck,
their scantlings are governed by the following minimum
section modulus.

$$Z = 10k \, Sh\ell_e^2$$

and above the lowest deck by

$$Z = 2.1 \, CkTS\ell_e \, \sqrt{D}$$

where all the parameters have already been defined. In both cases, the minimum web depth is 2.5 times the depth of the longitudinals. Suitable connections at top and bottom are to be provided to the web frames (3.3.1 and Table 7.3.1).

The minimum depth of the center girder d_{DB} is

$$d_{DB} = 28 \, B + 205 \, \sqrt{T}$$

where B is the rule breadth and T the draft. d_{DB} should not be less than 650 mm (Part 4, Chapter 1, 8.3.1). A greater depth is recommended in way of large engine rooms when the variation in draft between light and loaded conditions is considerable (4.1.1).

The minimum center girder thickness t is

$$t = (0.008 \, d_{DB} + 4) \, \sqrt{k}$$

t should not be less than 6 mm (Part 4, Chapter 1, 8.3.1).

In machinery spaces adjacent to the after peak, the double bottom is to be transversely framed. Elsewhere, transverse or longitudinal framing may be adopted, but for ships exceeding 120 m in length and for ships strengthened for heavy cargos, longitudinal framing is in general to be used (4.1.2 and Part 4, Chapter 1, 8.2.1).

If the double bottom is transversely framed, plate floors are to be fitted at every frame in the engine room. In way of boilers, plate floors are to be fitted under the boiler bearers (4.1.3). If the double bottom is longitudinally framed, plate floors are to be fitted at every frame under the main engines and thrust bearings. Outboard of the engine seating floors may be fitted at alternate frames (4.1.4).

The scantlings of floors clear of the main engine seatings are generally to be as required in way of cargo

spaces. In way of engine seatings, the floor minimum thickness is given by:

$$t_3 = (10 + 1.5\ f)\ mm$$

where f is the engine factor given by $P/R\ell$. P is the power of one engine at maximum service speed in KW, R is the rev/min of engine at maximum service speed, and ℓ is the effective length of engine foundation plate in m required for bolting the engine to the seating. In determining ℓ, the thrust and gearcase seating is to be considered as a separate item (4.1.5, 6.2.1 and Table 7.6.1).

Sufficient fore and aft girders are to be arranged in way of the main machinery to effectively distribute its weight and to ensure adequate rigidity of the structure. In midship machinery spaces, these girders are to extend for the full length of the space and are to be carried aft to support the foremost shaft tunnel bearing. This extension beyond the after bulkhead of the machinery space is to be for at least three transverse frame spaces, aft of which the girders are to scarf into the structure. Forward of the engine room forward bulkhead, the girders are to be tapered off over three frame spaces and effectively scarfed into the structure. In machinery spaces situated at the aft end, the girders are to be carried as far aft as practicable and the ends effectively supported by web frames or transverses (4.1.6).

Outboard of the engines, side girders are to be arranged where practicable to line up with the side girders in adjacent cargo spaces (4.1.7).

Where the double bottom is longitudinally framed and transverse floors are fitted in way of the engine seatings as required by the rules, no additional longitudinal stiffening is required in way of the engines other than the main engine girders, provided that the spacing of girders does not exceed 1.5 times the normal spacing of longitudinals. Where this spacing of girders is exceeded, shell longitudinals are to be fitted, and these are to scarf into the longitudinal framing clear of the machinery spaces (4.1.8).

The minimum thickness t of inner bottom plating in engine rooms clear of engine seatings is

$$t = 0.0015\ \sqrt[4]{Ltk}\ (S + 660)\ mm$$

t should not be less than 7.0 mm (4.1.9). In way of engine seatings integral with the tank top, the minimum thickness as given by Table 7.6.1 is

$$t = (19 + 3.4 \; f) \; mm$$

if two girders are fitted and

$$t = (25 + 3.4 \; f) \; mm$$

if one girder is fitted. f is the engine factor already defined, and A is the area of top plate in cm^2 for one side of seat, given by:

$$A = (120 + 44.2f + 4.07f^2) \; cm^2$$

The main engine girder total thickness for the case where two girders are fitted is

$$t_1 + t_2 = (28 + 4.08f) \; mm$$

If one girder only is fitted, we have

$$t_1 = (15 + 4.08f) \; mm$$

In general, one single girder can be accepted when all the following conditions apply (Table 7.6.1): f < 1.84, P < 5900 KW and L < 100 m.

Where the height of inner bottom in the machinery space differs from that in adjacent spaces, continuity of longitudinal material is to be maintained by sloping the inner bottom over an adequate longitudinal extent. The knuckles in the plating are to be arranged close to plate floors (4.1.10).

A.3 Det norske Veritas

All the rule sections given in parenthesis are taken from Chapter II.

The height of the center girder is to be sufficient to give good access to all parts of the double bottom and it is not to be less than:

$$h = 600 + 9B\sqrt{d} \; mm$$

where B is the greatest moulded breadth in m and d the
mean moulded summer draft in m. In the engine room, the
height of the tank top above the keel should be 45% and
30% greater than the required center girder height,
respectively, with and without a sump in way of the main
engine (Section 10, A305).

The thickness of inner bottom plating in engine and
boiler room is not to be less than

$$t = \frac{(3.5 + 0.023L)\ \ (S + 0.8)}{\sqrt{f_1}}\ \ mm$$

where L is the rule length, s is the frame spacing in m
and f_1 a material factor depending on the material (f_1
is equal to unity for mild steel) (Section 10, B501).

The thickness of the center girder in the engine room
is not to be less than (Section 10, Table D103):

$$\frac{6.5 + 0.05L}{\sqrt{f_1}}$$

The thickness of side girders, floors and brackets
in the engine room is not to be less than (Section 10,
Table D103):

$$\frac{6 + 0.035L}{\sqrt{f_1}}$$

In general, side girders are to be fitted so that the
distance between the side and center girders or the margin
plate or between side girders does not exceed the following
values: 5 m in longitudinally stiffened double bottom and
4 m in transversely stiffened double bottom (Section 10,
D201).

Girders are to be fitted under the machinery extending
from the bottom to the engine-seating top plate. If the
engine bed plates are bolted directly to the inner bottom,
the thickness of plating under the engines is to be at
least twice the rule thickness of inner bottom plating.
At least one side girder is to be fitted outside the engine
seating girders (Section 10, D202).

In the engine room, if the double bottom is longitudinally stiffened, floors are to be fitted at every second side frame. Bracket floors are to be fitted at intermediate frames extending to the first ordinary side girder outside the engine seating. In way of thrust bearing and below pillars, additional strengthening is to be provided (Section 10, D301).

If the double bottom is transversely stiffened, floors are to be fitted at every frame. In way of thrust bearing and below pillars, additional strengthening is to be provided (Section 10, D304).

Verticals in the engine room are to have a depth not less than 200 ℓ_1 mm where ℓ_1 is the span of the girder in m (Section 12, D101).

Girder flanges are to have a thickness not less than 1/30 of the flanges width when the flange is symmetrical, and not less than 1/15 when the flange is asymmetrical. The total flange width in the engine room is not to be less than 35ℓ mm (Section 12, D102).

In the engine and boiler room, side verticals are normally to be fitted at every fifth frame. For diesel machinery with a large number of cylinders and for turbine machinery, every fourth side vertical is normally to be replaced by a bulkhead between the ship's side and the supports under the casing side from the bottom to the lowest continuous deck (Section 12, D105).

A.4 Bureau Veritas

The rules first consider the particular case of cargo ships. If the double bottom is transversely framed, the minimum thickness of strakes in the inner bottom in way of the engine room is

$$e = 0.75 \sqrt{L + 10T_2} + 1.5 \text{ mm (minimum 7 mm)}$$

and in way of the boiler compartment:

$$e = 0.75 \sqrt{L + 10T_2} + 3.5 \text{ mm (minimum 9 mm)}$$

where L is the rule length and T_2 the draft. If the ship is longitudinally framed, the thickness is that given above but reduced by 0.5 mm (6.33.11).

If the spacing E of stiffeners (floors or longitudinals in m) is greater (or smaller) than the basic spacing E_O, the values are to be increased (or reduced) by $20(E-E_O)/3$ (6.33.12). E_O is the rule frame spacing in m, given by (6.12.11)

$$E_O = 0.72 \left(\frac{L}{100}\right)^{1/4}$$

In no case, should the thickness of the inner bottom plating be less than:

transverse framing: $e = 5.26 \ E \ \sqrt{h}$

longitudinal framing: $e = 4.45 \ E \ \sqrt{h}$

where $h = C_1 - H_D$, C_1 is the depth of ship in m to the deck below the top of overflow and H_D is the double-bottom depth in m (6.33.13).

In the engine and boiler space, the margin plate thickness is not to be less than that required for adjoining inner bottom plates (6.33.15).

The depth of the center girder is generally equal to (6.33.21):

$$b = 0.1 \ \sqrt{L}$$

and the thickness is not to be less than (6.33.22):

midship region: $e = 0.95 \ \sqrt{L + 10T_2}$ (minimum 7mm)

ends: $e = 0.80 \ \sqrt{L + 10T_2}$ (minimum 6mm)

The side girder thickness is not to be less than (6.33.24):

$$e = 0.7 \ \sqrt{L + 10T_2} + 1 \text{(minimum 7mm)}$$

Under the engine bed plates, additional girders are to be included. Under the thrust block, side girders are to be arranged to the satisfaction of the Head Office. Under the engine seatings, additional girders are to be arranged (6.33.25 and 6.33.75).

In the case of transverse framing, the thickness of plate floors is not to be less than

$$e = 0.7 \ \sqrt{L + 10T_2} + 1 \text{(minimum 7mm)}$$

In the case of longitudinal framing, the thickness is that given above increased by 10% (6.33.31).

In ships over 100 m in length or where the depth of floors exceeds 0.95m, floors are to be fitted with vertical stiffeners (6.33.84).

The number of side girders is to be such that the distance separating them from either another or the center girder or margin plate does not exceed 4.2m (6.33.74).

In transversely framed systems, plate floors are to be fitted at every frame within the machinery space, under the thrust block and under the boiler bearers. Under the main engines and auxiliaries, care must be taken to ensure that all the double-bottom items are well fitted (6.33.8).

In longitudinal systems, the floor spacing should be two frame spaces within the machinery and boiler spaces, one frame space in way of the main internal combustion engine and thrust blocks. In this case, bottom longitudinals may be reduced in scantlings (6.33.9).

In the engine room, web frames are to be generally fitted every fifth frame. The web frame depth is to be not less than twice that of the frame replaced and to have a section modulus not less than four times that of the frame (6.44.13).

The section modulus of frames is not to be less than that of the 'tweendeck or superstructure frame located just above nor than (6.43.21):

$$w = 3.5hE\ell^2$$

where h is the design load height in m, E is the frame spacing in m, and the span ℓ in m is to be measured between the level of the top of floors or tank top and the lowermost deck line. The loading height h is given by (6.42.11.):

$$h = h_s + 0.4\ h_o$$

with $h_o = 8.143 - 0.714\ (\frac{300-L}{100})^{3/2}$ if $L \leq 300$

$h_o = 8.143$ if $L > 300$

$h_s = 0.6\ \frac{b^2}{\ell_1}$ where the frame is partly immersed

$h_s = b - 0.4\ell_1$ when the frame is entirely immersed

ℓ_1 and b_1 are vertical distances in m, measured between the intersection of the side shell, with the extension of the top of floor or the inner bottom plating, and respectively the lower deckline side and the rule waterline.

If web frames are not provided, the section modulus should not be less than the value obtained from the formula given above increased by 15% where the engine room is not aft, or 40% if the engine room is aft (6.43.51).

If the particular case of tankers, the rules indicate that attention should specially be directed to the rigidity of the framing in the machinery space, particularly when high-power diesel engines are used (8.51.23).

In the double bottom, floors are to be provided at each frame, and girders in line with the bottom longitudinals in the cargo tank space are also to be provided. Where necessary, additional girders extending over a few frames are to be provided every two or three longitudinals, so as to ensure structure continuity (3.54.21).

The thickness of the inner bottom plating is not to be less than (8.54.22):

$$e = 0.75 \sqrt{L + 10T} + 2.5$$

The thickness derived from the formula is not to exceed 17 mm unless the overflow depth of the double bottom tanks justifies a greater thickness.

The thickness of ordinary floors is not to be less than (8.54.23):

$$e = 0.7 \sqrt{L + 10T} + 1$$

and it need not exceed 16 mm.

The thickness of watertight floors may be taken equal to that of ordinary floors increased by 1.5 mm, provided there are stiffeners spaced about 0.76 m and having a section modulus at least equal to

$$w = 6.5 \, H_D{}^2 \, h$$

where H_D is the double bottom depth in m and h is the distance in m between the tank top and the top of the overflow (8.54.24).

Girders, floors and inner bottom plating are to be suitably strengthened in way of the engines, reduction gears, thrust blocks, etc. (8.54.25).

As a rule, on the side shell plating, web frames are to be provided four frame spaces apart. The section modulus of web frames is not to be less than

$$w = 6E\ell^2 d$$

where ℓ is the span of web frames in m measured between flats or between the lower flat and the tank top, and d is the vertical distance in m between midspan and the main deck (8.54.31 and 32).

Where the side shell is framed transversely, the scantlings of the stringers are such that the section modulus is not to be less than

$$w = 10E^2 bd$$

where E is the web frame spacing in m, b is the width supported by the stringers in m and d is the vertical distance in m from the center of the area supported by the stringer to the deckline at side, without this distance being taken less than 4 (8.54.34 and 8.53.24).

A.5 *Germanischer Lloyd*

The rule sections mentioned below are taken from Chapter 2 (Construction Rules for Hull).

The rules specify that lightening holes in way of the engine foundation are to be kept as small as possible while keeping good accessibility. Where necessary, the edges of lightening holes are to be strengthened by means of face bars or the plate panels are to be stiffened (8.C.2.1.1).

Local strengthenings are to be provided beside the following minimum requirements according to the construction and the local conditions (8.C.2.1.2).

Plate floors are to be fitted at every frame. The minimum floor thickness in compartments other than machinery compartments is given by (8.B.7.2.1).

$$t = h/100 - 1.0 \text{ mm for } h \leq 1200 \text{ mm}$$

$$t = h/120 + 1.0 \text{ mm for } h > 1200$$

t need not exceed 16.0 mm. h is the depth of center girder in mm and its minimum value is given by (8.B.2.2).

$$h = 350 + 0.45B \text{ mm} > 600 \text{ mm}$$

where b is the greatest moulded breadth.

In machinery spaces, the floor thickness as given by the above expressions is to be strengthened as follows:

$$3.6 + N/500 \text{ (per cent)}$$

minimum 5%, maximum 15%, where N is the single engine output in KW (8.C.2.2).

At least one side girder shall be fitted in the engine room. The distance of the side girders from each other and from center girder and margin plate respectively shall not be greater than 1.8 m in the engine room within the breadth of engine seatings (8.B.3.1).

The thickness of side girders under an engine foundation top plate inserted into the inner bottom is to be similar to the thickness of side girders above the inner bottom as defined below (8.C.2.3.1).

Side girders with the thickness of longitudinal girders are to be fitted under the foundation girders in full height of the double bottom. Where two side girders are fitted on either side of the engine, one may be a half-height girder under the inner bottom for engines up to 3000 KW (8.C.2.3.2).

Side girders under foundation girders are to be extended into the adjacent spaces and to be connected to the bottom structure. This extension aft and forward of the engine room bulkheads shall be 2-4 frame spaces if practicable (with machinery aft, only forward of the engine room) (8.C.2.3.3).

No center girder is required in way of the engine seating but intercostal docking profiles are to be fitted instead. The sectional area of the docking profiles is not to be less than (8.C.2.3.4 and 8.C.1.4).

$$f = 10 + 0.2L$$

where L is the rule length. Docking profiles are not required where a bar keel is fitted. Brackets connecting the floor plates to the bar keel are to be fitted on either side of the floors.

Between the foundation girders, the thickness of the inner bottom plating is to be increased by 2 mm over the value it has in other locations. The strengthened plate is to be extended beyond the engine seating by three to five frame spacings (8.C.2.4).

Regarding the design of engine seatings, the rules give some recommendations which apply to low-speed engines. Seatings for medium and high-speed engines as well as for turbines must be specially considered (8.C.3.1.1).

The rigidity of the engine seating and the surrounding bottom structure must be adequate to keep the deformations of the system due to the loads within the permissible limits. In special cases, evidences may be required of deformations and streses (8.C.3.1.2).

Regarding the foundation of diesel engines, the rules offer the following guidance (8.C.3.1.2):

At the draught resulting in the maximum deflection in way of the foundation, the deflection of two stroke, crosshead engines including foundation ought to be less than 1 mm over the length of the engine. In addition to the deflection of engine and foundation, the crank web deflections by which the admissible engine deflection may be limited to values much less than 1 mm have to be considered as well. For medium-speed and high-speed engines, not only the deflections of crank webs have to be taken into account, but for assuring trouble-free bearing conditions of the crank shaft the bending deflections of the engine is to be limited.

Due regard is to be paid, at the initial design stage, to a good transmission of forces in transverse and longitudinal direction (8.C.3.1.3).

The foundation bolts for fastening the engine at the seating shall be spaced no more than 3 d apart from the longitudinal foundation girder. Where the distance of the foundation bolts from the longitudinal foundation girder is greater, proof of equivalence is to be provided. d is the diameter of the foundation bolts (8.C.3.1.4).

In the whole speed range of main propulsion installations for continuous service, resonance vibrations with inadmissible vibration amplitudes must not occur; if necessary structural variations have to be provided for avoiding resonance frequencies. Otherwise, a barred speed range has to be fixed. Within a range of -10% to +5% related to the rated speed no barred speed range is permitted. The Society reserves the right to demand in special cases a proof of vibration-free service (8.C.2.1.5).

The thickness of the longitudinal girders above the inner bottom is not to be less than:

$$t = \sqrt{N/15} + 6 \text{ mm for } N < 1500 \text{ KW}$$

$$t = N/750 + 14 \text{ mm for } 1500 \leq N \quad 7500 \text{ KW}$$

$$t = N/1875 + 20 \text{ mm for } N > 7500 \text{ NW}$$

where N is the single engine output in KW (8.C.3.2.1).

Where two longitudinal girders are fitted on either side of the engine, their thickness as given by the formulas above may be reduced by 4 mm (8.C.3.2.2).

The sizes of the top plate (width and thickness) shall be sufficient to attain efficient attachment and seating of the engine and depending on seating height and type of engine adequate transverse rigidity.

The thickness of the top plate shall approximately be equal to the diameter of the fitted-in bolts. The cross-sectional area of the top plate is not to be less than:

$$F_T = N/15 + 30 \text{ cm}^2 \text{ for } N \leq 750 \text{ KW}$$

$$F_T = N/75 + 70 \text{ cm}^2 \text{ for } N > 750 \text{ KW}$$

Where twin engines are fitted, a continuous top plate is to be arranged in general if the engines are coupled to one propeller shaft (8.C.3.2.3).

The longitudinal girders of the engine seating are to be supported transversely by means of web frames or wing bulkheads (8.C.3.2.4).

Top plates are preferably to be connected to longitudinal and transverse girders thicker than approximately 15 mm by means of a double bevel butt joint (8.C.3.2.5).

In the engine and boiler room, web frames are to be fitted. Generally, they should extend up to the uppermost continuous deck. Where the depth is 4 m, the web frames are to be spaced 3.5 m apart on an average, where the depth if 14 m, they are to be spaced 4.5 m apart on an average (9.A.8.1.1).

For combustion engines up to about 400 kW, the web frames shall generally be fitted at the forward and aft ends of the engine. For combustion engines of 400 to 1500 kW, an additional web frame is to be provided at half length of the engine, and for engines with higher outputs, at least two further web frames are to be provided (9.A.8.1.2).

Where combustion engines are fitted aft, stringers spaced 2.6 m apart are to be fitted in the engine room, in alignment with the stringers in the after peak, if any, or else, the main frames are to be adequately strengthened. The scantlings of the stringers shall be similar to those of the web frame. At least one stringer is required where the depth up to the lowest deck is less than 4 m (9.A.8.1.3).

The section modulus of the web frames is not to be less than (9.A.8.2):

$$w = k \; 0.8 \; e \; \ell^2 \; P_s \; cm^3$$

where k is a material factor (equal to unity for ordinary hull structural steel), e is the web frames spacing in m, ℓ is the span and p_s is the load in KN/m^2 on the ships side.

The moment of inertia of the web frame is not to be less than:

$$J = H(4.5H - 3.75) \; c \; 10^2 \; cm^4$$

where $3m \leq H \leq 10m$

$$J = H(7.25H - 31) \; c \; 10^2 \; cm^4$$

where $H > 10m$

$$c = 1 + (H_u - 4) \; 0.07$$

where H is the rule depth and H_u is the depth measured to the lowest deck in m. The scantlings of the webs (depth h and thickness t) are to be calculated as follows:

$$h = \; 50.H \; mm > 250 \; mm$$

$$t = h/(32 + 0.03h) \; mm > 8.0 \; mm$$

Ships with a depth of less than 3 m are to have web frames with web scantlings not less than 250 x 8 mm and a minimum face sectional area of 12 m^2.

APPENDIX B: BEARING REACTIONS COMPUTER PROGRAM

The Boston Naval Shipyard (BNS) Computer Code [56] is used to compute the bearing reactions of a ship's propulsion shafting system. Bearing reactions are computed when all the bearings are on a straight line. Alternatively, the reactions for other than the straight line conditions can be calculated using a matrix of influence numbers which are computed by the BNS Computer Code. Thus, the effect on the bearing reactions magnitude of raising or lowering any particular bearing can be found using the influence number table. The BNS Computer Code also computes at given points of the shafting system, the shear force, bending moment, slope and transverse deflection value. The details of the theoretical procedure for the Shafting calculation can be found in Reference [56].

In order to prepare the input data for the BNS Computer Code, the shafting system must be divided into a number of uniform sections. For each section of the shaft the following must be specified:

- length of shaft section

- outside diameter

- second diameter

- material

The computer output consists of the following:

- shear force

- bending moment

- slope

- transverse deflection

Figure 32 illustrates the usefulness of the computer code ANALYSIS for hull-machinery compatibility studies. The engine-room double-bottom flexibility matrix $\mathcal{F}$ is computed by the ANALYSIS code. The matrix $\underset{\sim}{I}$ of influence coefficients is computed using the Boston Naval Shipyard Computer Code [56].

APPENDIX C: COMPUTER CODE ANALYSIS

The deformation of the engine room structure at any point has the following contributions:

- deformation of hull girder

- deformation of 2-D frame

- deformation of double-bottom grillage structure

The software computer code developed consists of the following modules:

- GIRDER for hull girder analysis

- SPACE for 2-D frame and grillage analysis of double bottom

The computer code ANALYSIS contains the modules GIRDER and SPACE. Any of the modules can be executed any number of times and in the desired order. The modules are written in Fortran IV.

The input parameters to the ANALYSIS Computer code are described in the following. A computer listing is provided thereafter.

ANALYSIS can accept input in any dimensions. However, the input should be dimensionally compatible and the same dimensions should be used for the entire package.

Below the main heading, a 'check for dimensions' is printed out. The user should make sure the dimensions are compatible:

ALL INPUT IS FORMAT FREE EXCEPT WHERE GIVEN.

Card 1: DLEN, DWET

DLEN: dimension for length (4 characters)   - $\begin{cases} \text{FEET} \\ \text{INCH} \\ \text{METER} \end{cases}$

DWET: dimension for weight (5 characters)   - $\begin{cases} \text{POUND} \\ \text{TON} \\ \text{KG} \end{cases}$

CARD 2: PRM

    PRM: Program module to be executed

        'GIRDER' for girder analysis
        'FRAME' for frame analysis
        'GRILLAGE' for grillage analysis

Depending on the value of PRM - go to the appropriate section for input parameters.

GIRDER - Input Parameters

CARD 1: ALBP, PMEA, PMEF, WTLS, CGLS

    ALBP: Length between perpendiculars
    PMEA: Aft end of parallel middle body from AP
    PMEF: Forward end of parallel middle body from AP
    WTLS: Distributed hull weight
    CGLS: Distributed hull weight C.C. from AP

CARD 2: YEM, GEE, FØF

    YEM: Young modulus of elasticity
    GEE: Shear modulus of elasticity
    FØF: Form factor for shear deflection

CARD 3: THM, TMLC

    THM: Thrust moment
    TMLC: Location of thrust moment from AP

CARD 4: NBP

    NBP: No of points for which buoyancy values are
          input - max 20

If there is a discontinuity in buoyancy curve input 2 points for the same location

eg:
TREATMENT OF DISCONTINUITY IN BUOYANCY CURVE

x = input 2 points here.
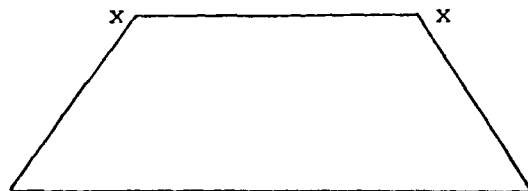
Figure 48

CARD 5:   BPL

    BPL: Distance of buoyancy points from AP

CARD 6:   BPV

    BPV: Buoyancy per unit length value at each point

CARD 7:   NS

    NS: No. of semi-distributed weight items - max 50

CARD 8:   NU, ANL1, ANL2, ANL3, ANL4, ANL5, WL, GLL, DLL, AL -
    (formatted input)

    NU: Serial no. of weight item - I2
ANL1, ANL2, ANL3, ANL4, ANL5 - alphanumeric description
    of item: 20 characters
    WL: Weight of item F8.2
    GLL: Distance of CG of weight item from AP F8.2
    DLL: Length over which weight item is distributed F8.2
    AL: Distance of aft end of weight from AP F8.2

SPACE - Input Parameters

Preparation of input data for this program should be accomplished in the following sequence:

1. Sketch the structure and number the joints and members as indicated in Fig.49, remembering to observe the geometry of the structure in order to determine the joint sequence that will keep the half-band width of the stiffness matrix as narrow as possible.

2. Establish the reference coordinate system and label the joints with the proper coordinate values.

3. Define the different load cases to be considered.

4. With the aid of items 1 through 3 above, prepare data cards according to the formats indicated in the following descriptions.

5. The units should be dimensionally compatible, i.e. for eg:
    Length: feet
    Area: $feet^2$
    Inertia: $feet^4$
    Modulus of elasticity: $KIPS/feet^2$
    Load: KIPS
    Moment: KIPS - feet
    Distributed load: KIPS/feet

STRUCTURE DATA

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'S' for structure card	A1
2-4	M	Number of members in structure.	I3
5-7	NJ	Number of joints in structure.	I3
8-10	NL	Number of loading conditions for this problem.	I3
11-13	MUD	The half-band width of the stiffness matrix (estimated).	I3
14-23	EN	Global modulus of elasticity for this problem	F10.0
24-75	—	Blank.	52X
76-80	JOBNO	Job identification number (any one - to five-digit number).	76-80

JOINT DATA

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'J' for joint data card.	A1
2-4	J	The joint number of this joint.	I3
5	IXT	X-coordinate translational restraint of this joint. Leave blank if this joint is unrestrained in X-coordinate direction. Place '1' in this column if this joint is restrained in X-coordinate direction.	I1
6	IYT	Same as IXT above, except in Y-coordinate direction.	I1
7	IZT	Same as IXT above, except in Z-coordinate direction.	I1
8	IXR	Rotational restraint of this joint in X-coordinate direction. Leave blank if joint is.	I1

unrestrained in X-coordinate
direction. Place '1' in this
column if this joint is re-
strained in X-coordinate di-
rection.
Note: For space truss problems,
place '1' in this column.

9	IYR	Same as IXR above, except in Y-coordinate direction.	I1
10	IZR	Same as IXR above, except in Z-coordinate direction	I1
11-20	XCOOR	X-coordinate of this joint.	F10.2
21-30	YCOOR	Y-coordinate of this joint.	F10.2
31-40	ZCOOR	Z-coordinate of this joint.	F10.2
41-80	———	Blank.	——

MEMBER DATA

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'M' for member data card.	A1
2-4	I	Member number.	I3
5-7	J	Joint number of end j of member.	I3
8-10	K	Joint number of end k of member.	I3
11	MT	Member type. Leave column 11 blank if space frame member, place '1' in column 11 if space truss member.	I1
12-20	QIX	Moment of inertia about member X-axis.	F9.2
21-30	QIY	Moment of inertia about member Y-axis.	F10.0
30-40	QIZ	Moment of inertia about member Z-axis.	F10.0
41-50	QA	Cross-sectional area of member.	F10.0
51-60	G	Shear modulus of elasticity.	F10.0

61-70	SI	Angle of roll Ψ in degrees.	F10.0
71	ISI	If the angle of roll is specified for rotation YZX, leave this column blank. If specified for rotation ZYX, place '1' in this column.	I1
72-80	E	Modulus of elasticity of this member if different from global modulus assigned on structure data card. If same as global modulus, leave this field blank.	F9.0

MEMBER LOAD DATA

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'L' for load data card.	A1
2-4	IB1	Member number of this number.	I3
5	IB2	Plane of loading. If the load lies in the member axis x_m-y_m plane, leave this column blank. (See Fig. 51, load P.) If the load lies in the member axis x_m-z_m plane, place '1' in this column. (See Fig. 51, load Q.)	I1
6-10	——	Blank.	5X
11-20	AB1	Value of load/length if uniform load is specified; load if a concentrated load is specified.	F10.3
21-30	AB2	Distance from joint j of member to beginning of load.	F10.3
31-40	AB3	Distance from joint j of member to termination of load.	F10.3

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
41-50	AB4	The angle the load makes with a normal line in degrees - i.e., α in Fig. 51.	F10.3
51-60	AB5	Blank (used in reading joint load data).	F10.3
61-70	AB6	Blank (used in reading joint load data).	F10.3
71-80	---	Blank.	-----

JOINT LOAD DATA

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'P' for joint load card.	A1
2-4	IBI	The joint number.	I3
5	IB2	Blank (used in reading member load data).	I3
6-10	---	Blank.	5X
11-20	AB1	Applied force in X-coordinate direction at this joint	F10.3
21-30	AB2	Same as AB1 above, except in Y-coordinate direction.	F10.3
31-40	AB3	Same as AB1 above, except in Z-coordinate direction.	F10.3
41-50	AB4	Applied moment about N-axis at this joint.	F10.3
51-60	AB5	Applied moment about Y-axis at this joint.	F10.3
61-70	AB6	Applied moment about Z-axis at this joint.	F10.3
71-80	---	Blank.	-----

DUMMY LOAD DIVIDER CARD

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'N' to indicate termination of this loading condition and the beginning of a new loading condition. The letter 'E' to terminate the last loading condition for this problem.	Al
2-80	----	Blank	--

PROGRAM TERMINATION CARD

Card Columns	Identifier Name Used in Program	Data Description	FORTRAN Format
1	KODE	The letter 'Q' to tell the program to quit execution. This is the last card in the data deck.	Al
2-80	----	Blank	----

NUMBERING SCHEME FOR SPACE FRAME
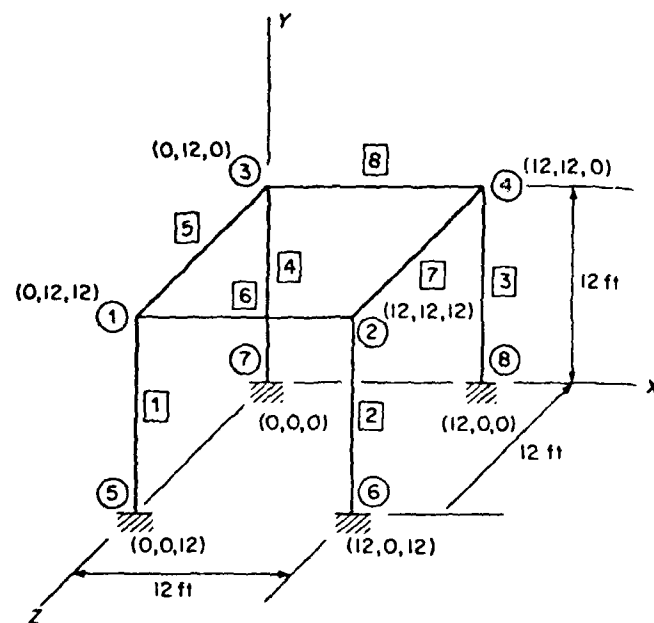
FIGURE 49

DISPLACEMENT DESIGNATION SEQUENCE FOR SPACE FRAME JOINT
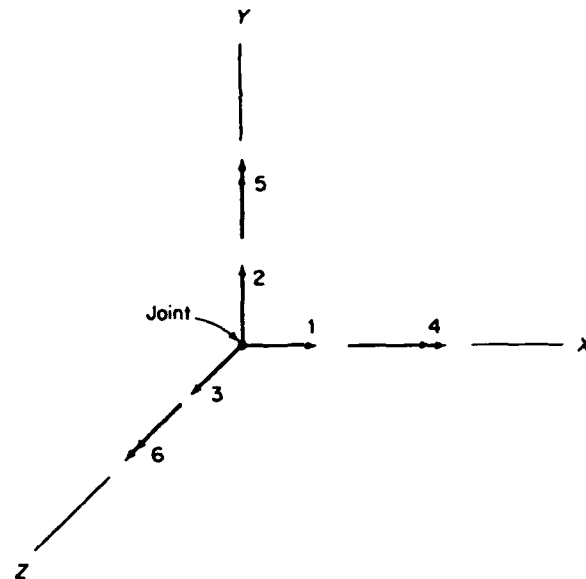
FIGURE 50

SIGN CONVENTION FOR MEMBER LOADS

FIGURE 51

APPENDIX D: STRUCTURAL ANALYSIS OF ENGINE ROOM COMPUTER CODE LISTING
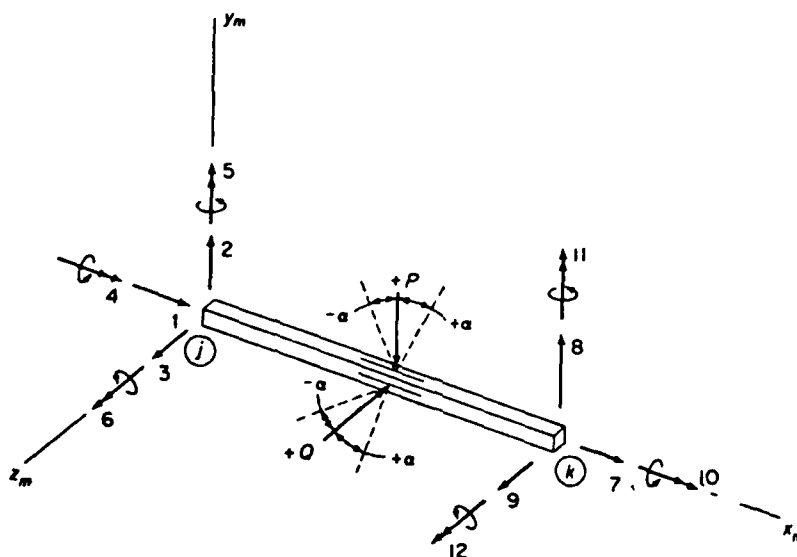
```
100  C *********************************************************
200  C MAIN PROGRAM FOR STRUCTURAL ANALYSIS OF ENGINE ROOM
300  C CONSISTS OF THREE SECTIONS:  1) GIRDER ANALYSIS
400  C                              2) FRAME ANALYSIS
500  C                              3) THRUST BEARING FOUNDATION ANALYSIS
600  C                              4) GRILLAGE ANALYSIS
700  C *********************************************************
800         CHARACTER*8 PRM,DIEM
900         CHARACTER*5 DWET
1000        CEM=CB*(1/PEM
1100        READ(5,1010)DIEM,DWET
1200        WRITE(6,105)
1300        READ(5,1010)END=1111)PPP
1800 1000   READ(5,1010)DIEM,DWET,DIEM,DIFM,DWET,DIEM,DIEM,DWET,DIEM
1500        IF(PRM.EC.'GIPD')CALL GIRDER
1600        IF(PRM.EC.'FRAM')CALL FRAME
1700        IF(PRM.EC.'GRIL')CALL SPACE(PRM)
1800        IF(PRM.EC.'RCTF.CR.PRM.EC.'SHEA'.CR.PRM.EQ.'BEND') CALL TBSTIFF
1900        IF(PRM.NE.'GIRD'.AND.PRM.NE.'FRAM'.AND.PRM.NE.'RCTA'.AND.
2000        )PRM.NE.'SHEA'.AND.PRM.NE.'BEND'.AND.PRM.NE.'GRIL')THEN
2100        TYPE 1112
2200        GO TO 1111
2300        END IF
2400        GO TO 1000
2500  C *********************************************************
2600  C FORMAT SECTION
2700  C *********************************************************
2800 1001   FORMAT(A8)
2900 1010   FORMAT(A8,2A)
2910 1005   FORMAT(5X,'---- ANALYSIS MODULE NOT CORRECT',//,5X,
2920        1'---- PROGID BE --- CREEP OF FRAME OR FUNCTION',//,5X,(2X)
3010 1013   FORMAT(7X,//,132('*'),/,132('*'),/,130X,'*',15X,30X,'*',
3100        1/,1X,35X,'ANALYSIS OF ENGINE ROOM STRUCTURE USING SIMPLIFIED',
3200        2'METHODS',57X,'*',/,135X,'*',//,5X,'DEVELCPF',/,135X,'*',
3300        3. DEPARTMENT OF OCEAN ENGINEERING JUNE 1980',55X,'*',/,15X,'*',
3600        4130X,'*',//,1X,130X,'*',/,132('*'),/,132('*'))
3700 1014   FORMAT(15(//),15X,CHECK FOR DIMENSIONS IN THIS EXECUTION',//,
3800        1,15X,5CRCCE',//,15X,'DEL ---',//,15X,'WEIGHT',//,15X,
3900        2,15X,'FORCE',//,15X,'CHEMI',//,15X,'APEA',//,15X,30X 2,//,15X
4000        3,'THRUTN',//,15X,'PODULUS',//,15X,'A ',//,5X,'A ',//)
4100  C *********************************************************
4200 1111   STOP
4300        END
4400  C *********************************************************
4500  C END OF MAIN PROGRAM
4600  C *********************************************************
4700  C
4800        SUBROUTINE GIRDER
4900  C *********************************************************
5000  C ANALYTICAL METHODS PACKAGE FOR ANALYSIS OF ENGINE ROOM STRUCTURE
5100  C *********************************************************
5200  C PART 1 FULL GIRDER ANALYSIS
5300  C *********************************************************
5400  C INPUT AND ECHO SECTION
5500  C *********************************************************
5600        DIMENSION PPL(20),EPN(20),EMG(20),EPL1(50),ANI2(50),ANI3(50),
5700        1WLE(50),GIL(50),DIL(50),AL(50),F(50),RP(60),PWL4(50),
5800        3SEF(100),BD(100),BCW(1C1),W(100),BW(100),SI(100),DLTA(100),
5900        3SEF(100),TDF(100),ICR(40),AFIS(50)
6000        WRITE(6,357)
6100        READ(5,*)ALPP,PFEA,PPEF,WTLS,CGLS
```

```
6200        READ(5,*)PM,CEE,FCF
6300        IF(PCF.EC.0.0)PCF=1.0
6400        READ(6,372)ALSD,PFEA,PWEF,WTLS,CGLS
6500        READ(5,*)PM,TBLC
6600        WRITE(6,3C3)TMP,TBLC
6700        WRITE(6,380)PPE,GEE
6800        READ(5,*)NBD
6900        READ(5,*)(PPL(I),I=1,NBP)
7000        READ(5,*)(BPV(I),I=1,NBP)
7100        WRITE(6,304)
7200        WRITE(6,325)(PPL(I),BPV(I),I=1,NBP)
7300        READ(5,*)NS
7400        READ(5,2C6)(NU(J),PML1(J),AML1(J),API2(J),API3(J),ANI4(J),ANI5(J),PL(J),
7500        1GL(J),DIL(J),AL(J),J=1,NS)
7600        WRITE(6,307)
7700        WRITE(6,3C8)(NU(J),PML1(J),AML1(J),API2(J),API3(J),API4(J),API5(J),VL(J),
7800        1CI(J),DIL(J),AL(J),J=1,NS)
7900  C *********************************************************
F10  C DETERMINE WEIGHT DISTRIBUTION
820  C *********************************************************
830         PBL=PPE-PFEA
840         POL=ALPP-PWEF
F40         VI=5.0-PL/ALPP
850         PC=WTS-VI*C.25/ALPP
860         WPT=WTLS-CGLS
870         REC=WTLS-PC-EBL
880         REC2=WPT-PC(2*P)*(PPL*0.5+FWEF)
900         RPS2=6.*RPC2-5C-2.*(PPEA-2)-PC*(FOL*2)-PC*F*PEF*FCF*3.0
910         C1=(2.-F(II*2))*(1.0*POL*PWEF)*(PPEA*FCL)
920         FCPD=(P*S2+BPS1*PPEA))/C1
930         ACRD=(F*S1-ICPF*FOL)/PWFA
940  C *********************************************************
950  C AREA UNDER WEIGHT CURVE AND DISTRIBUTION MAPPING
960  C *********************************************************
970         CIL=AIBP/10C.
980         X(1)=C.C
990         WD(1)=0.0
1000        CPP(1)=ACRD/PC
1010        DA=(FC-ACRD)/PFEA
1020        DF=(FCRE-PCW)/FCL
1030        BLK1=PWEA-(1C*D*PC)*0.5
1040        BLK2=DRI-EC*BLK1
1050        DC WC1 1=2,101
1060        X(1)=X(1-1)*CIL
1070  4C1   WD(I)=WD(I-1)*(AGRD*C5*C.5*X(1))
1090        DWPL(1)=F(I-1)*(CIL-EP/PC)
1100        GC TC 6
1110  4C5   IF(X(I)-PWEF)404,4C4,405
1120  4C6   WD(I)=X(I)-PFEA)*EC*BLK1
1300        DWPI(I)=1
1400  11900  GC TC 6C6
1500  4C6   VAR=X(I)-PWEF
1600        WD(I)=VAR*(FC*DP-C.5*VAR)*BLK2
1700        DWPL(I)=PWEF(I-1)*(CIL-TF/PC)
1800  406   DC WC7 I=1,NS
1900        IF(X(I)-ITAL(J))GC TC 4C7
1200        IVTV(1)=A(C,J)
1210        IF(KYT-DIL(J))4C8,4C9,4C9
1220  4C6   PHS1=2.-W(J)*PLL(J)
```

```
18400      CJ(I)=0.0
18500      CJ(I)=0.0
18600      EJ(I)=0.0
18700      PJ(I)=0.0
18800      DK(I)=0.0
18900      SK(I)=0.0
19000      PK(I)=0.0
19100      DK(I)=0.0
19200      FK(I)=0.0
19300  235 OK(I)=0.0
19400      DC 235 I=NB,NEND
19500  235 STIF(I)=0.0
19600  8021 READ(IN,12B)KCDE,IB1,IB2,AB1,AB2,AB3,AB4,AB5,AB6
19700  12B FCRMAT(A1,I3,I1,5X,6F10.3)
19800      IF(KCDE.EQ.IC6)GO TO 8006
19900      IF(KKCDE.EQ.IC5)GC TC 8732
20000      IF(KCDE.EC.IC5.OR.KCDE.EQ.IC7)GO TO 8008
20100      GC TC 13
20200  8008 IF(L39.GT.C)GC TO 393
20300      WRITE(IT,518)IJJ
20400  918 FCRMAT(1H1,3CX,'MEMBER LCAD DATA FCR LCADING NUMBER',I4,///,25X,
20500      1'MEMBER',4X,'LOAD',7X,'X BEGIN',5X,'X END',5X,'ALPHA',3X,'TYPE'/,
20600      2132('-'))
20700  393 YL=IB1
20800      L39=1
20900      FT=BT2(KL)
21000      W=AB1
21100      YX1=AB2
21200      YY2=AB3
21300      NIPHA=AB4
21400      WRITE(IT,512)KL,AB1,AE2,AB3,AB4,AB5,IB2
21500  912 FCRMAT(26X,I3,2X,F10.2,3E11.2,I4)
21600      IF(NIPHA)8009,8010,8011
21700  8010 CCSA=1.
21800      SINA=0.
21900      GC TC 8020
22000  8009 IF(ALPHA+9C.)8012,8013,8012
22100  8013 CCSA=0.
22200      SINA=-1.
22300      GC TC 8020
22400  8011 IF(AIPHA-9C.)8012,8015,8012
22500  8015 CCSA=0.
22600      SINA=1.
22700      GC TC 8020
22800  8012 AIPHA=ALPHA*3.14159/180.
22900      CCSA=CCS(ALPHA)
23000      SINA=SIN(ALPHA)
23100  8020 QZ=QI11KI)
23200      IF(M1.EC.0)CALL FIV(W,XX1,XX2,OL,SINA,CCSA,P11,P22,P33,F44,P55,P'
23300      1)
23400      IF(M1.GT.0)CALL FIX3(W,XX1,XX2,OL,SINA,COSA,P11,F22,F33,F44,P55,
23500      1P66)
23600      TJ(KI)=TJ(KI)+P11
23700      DK(KI)=CK(KI)+P44
23800      IF(IB2.EQ.1)GC TC 7681
23900      PJ(KI)=RJ(KI)+P22
24000      PJ(KI)=SJ(KI)+P33
24100      SK(KI)=SK(KI)+P55
24200      QK(KI)=QK(KI)+P66
24300      GC TC 8'21
24400  7681 CJ(KI)=GJ(KI)+P22
```

```
24500      EJ(KI)=EJ(KI)+P33
24600      HK(KI)=HK(KI)+P55
24700      FK(KL)=FK(KL)+P66
24800      GC TO 8'21
24900  873: IF(L3B.CT.0)GO TO 37
25000      WRITE(IT,9685)IJJ
25100  9685 FCRMAT(1P1,32X,'JCINT LCAD DATA FCR LCADING',I4,///,1CX,'JOINT',5X
25200      1,'X-FORCE',5X,'Y-FORCE',5X,'Z-FORCE',5X,'X-MCMENT',4X,'Y-MCMENT',
25300      2 24X,'Z-MCMENT',/,132('-'),/,)
25400  37  J=IB1
25500      L3B=1
25600      F11=IXD(J)+NN
25700      K22=IYD(J)+NN
25800      K33=IZD(J)+NN
25900      K44=IRX(J)+NN
26000      K55=IPY(J)+NN
26100      K66=IPZ(J)+NN
26200      IF(K11.GT.N1)FX1)K11=NDIM1
26300      IF(K22.GT.NDIM1)K22=NDIM1
26400      IF(K33.GT.NDIM1)K33=NDIM1
26500      IF(K44.GT.NDIM1)K44=NDIM1
26600      IF(K55.GT.NDIM1)K55=NDIM1
26700      IF(K66.GT.NDIM1)K66=NDIM1
26800      STIF(K11)=STIF(K11)+AB1
26900      STIF(K22)=STIF(K22)+AB2
27000      STIF(K33)=STIF(K33)+AB3
27100      STIF(K44)=STIF(K44)+AB4
27200      STIF(K55)=STIF(K55)+AB5
27300      STIF(K66)=STIF(K66)+AB6
27400      WRITE(IT,93)J,AB1,AB2,AB3,AB4,AB5,AB6
27500  93  FCRMAT(1H ,I2X,I3,6F12.2)
27600      GC TC 8'21
27700  8006 CCNTINUE
27800      IF(L39.EQ.0)GO TO 25
27900      DC P031 LLL=1,M
28000      CALL POT(T,CX(LLL),CY(LLL),CZ(LLL),SI(LLL),IST(LLL))
28100      FEM(1)=TJ(LLL)
28200      FEM(2)=RJ(LLL)
28300      FEM(3)=SJ(LLL)
28400      FEM(4)=CJ(LLL)
28500      FEM(5)=EJ(LLL)
28600      FEM(6)=FJ(LLL)
28700      FEM(7)=UK(LLL)
28800      FEM(8)=SK(LLL)
28900      FEM(1C)=DK(LLL)
29000      FEM(11)=FK(LLL)
29100      FEM(12)=CK(LLL)
29200      DC 9997 I=1,12
29300      FEM(I)=0.0
29400      DC 9997 J=1,12
29500  9997 FEM(I)=FEM(I)+R(I,J)*FEM(J)
29600      DC 8291 I=1,12
29700  K1=K1(LLL,I)+NN
29800      IF(K1.GE.NDIM)K1=NDIM1
29900  8291 STIF(IF(11)=STIF(IF(11)-FEM(I)
30000  8031 CCNTINUE
30100  3C  CALL CHOLES(STIF,R,NN,C,IJJ,IT,6977)
30200      STIF(NDIM1)=C.0
30300      WRITE(IT,9C9)IJJ
30400  9C9 FCRMAT(1H1,3CX,'JCINT DEFLECTIONS AND ROTATIONS FCR LCADING',I4,
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944 FORMAT(' STORAGE AVAILABLE FOR STIFFNESS MATRIX AND ONE LOAD',
   4' VECTOR IS INSUFFICIENT',/,20X,'STORAGE REQUIRED=',I6,2X,
   2'STORAGE AVAILABLE=',I6)
    GO TO 977
    END
C***********************************************
    SUBROUTINE FIX3(W,P,OL,SINA,CCSA,P11,P22,P33,P44,P55,P66)
    P33=C.
    P66=0.
    C=B-A
    IF(ABS(C)-0.05)1,1,2
1   WU=W*COSA
    P11=W*SINA*(QL-P)/QL
    P44=W*SINA-P11
    GO TO 3
2   P44=W*SINA*(A-B)*C/(2.*OL)
    P11=W*SIN.C-P44
    WU=W*CCSA
    D=QL-C.5*(A+B)
    P22=WU*D/QL
    P55=WU-P22
    RETURN
    END
C***********************************************
    SUBROUTINE FIX(W,XX1,XX2,CL,SINA,CCSA,F11,F22,P33,P44,C55,P66)
    C=XX2-XX1
    IF(ABS(C)-.C5)1,1,2
1   WU=W*COSA
    P11=W*SINA*(QL-XX1)/QL
    P44=W*SINA-P11
    GO TO 3
2   WU=W*C*CCSA
    P44=W*SINA*C*(XX1-XX2)/(2.*OL)
    P11=W*SINA*C-P44
    D=CL-XX1/2.-XX2/2.
    TS1=WU/(24.*OL)
    TS2=24.*C*3/CL-6.*XX2*C*2/OL+3.-C**3/OL
    TS3=6.*C*C-24.*D*D
    P33=.*C*C-48.*D*D+24.*D*OL
    P66=-TS1*(TS2*TS3)
    TS1=TS1*6./OL
    TS2=12.*C-C-8.-D**3/OL+2.*XX2*C*C/OL-C**3/OL-C*C
    P22=TS1*TS2
    P55=WU-P22
    RETURN
    END
C***********************************************
C***********************************************
    SUBROUTINE ROTR(T,CX,CY,CZ,SI,ISI)
    DIMENSION T(12,12)
    REAL*8 ,D1,D2
    D1=22./7.
    D2=180.
    D1=D1/D2
    DO 99 I=1,12
```

```
1//,10X,'JOINT',5X,'X-DEFL',6X,'Y-DEFL',6X,'Z-DEFL',4X,'ROTA X-AX',
26X,'ROTA Y-AX',6X,'ROTA Z-AX',/,132(-.))
    DO 100 I=1,NJ
    K11=IXD(I)+NM
    K22=IYD(I)+NM
    K33=IZD(I)+NP
    K44=IRX(I)+NT
    K55=IRY(I)+NT
    K66=IRZ(I)+NP
    IF(K11.GT.NDIM)K11=NDIM1
    IF(K22.GT.NDIM)K22=NDIM1
    IF(K33.GT.NDIM)K33=NDIM1
    IF(K44.GT.NDIM)K44=NDIM1
    IF(K55.GT.NDIM)K55=NDIM1
    IF(K66.GT.NDIM)K66=NDIM1
150 WRITE(IT,915)I,STIF(K11),STIF(K22),STIF(K33),STIF(K44),STIF(K55),
   1STIF(K66)
910 FORMAT(IM ,12,I3,6E12.5)
    WRITE(IT,91)JJJ
911 FORMAT(18),85X,'FINAL MEMBER END ACTIONS LOADING',I8,//,2X,
   1'MEMBER  J Y-FCE  J Z-FCE  J X-MOM  J Y-MOM  K',
   2' X-FCE  K Y-FCE  K Z-FCE  K X-MOM  K Z-CP',/,132(-.))
    DO 125 IJK=1,5
    OI=OL+I(IJK)
    CALL PCTR(C,CX(IJK),CY(IJK),CZ(IJK),SI(IJK),ISI(IJK))
    CALL MEMBER(T,AK,AKT,F(IJK),OIX(IJF),OIX(IJK),CIX(IJK),OA(IJK),
   1G(IJK),CL,MT)
    DN(II)=0.0
    DO 117 II=1,12
    DN(II)=0.0
    DO 117 J=1,12
    K11=K*(IJK,J)+NM
    IF(K11.GE.NDIM)K11=NDIM1
117 DN(II)=DN(II)+T(II,J)*STIF(K11)
    DO 130 J=1,12
    FEM(J)=C.
    DO 130 K=1,12
130 FEM(J)=FEM(J)+AK(J,K)*DN(K)
    AFM(1)=FEM(1)+TJ(IJK)
    AFM(2)=FEM(2)+RJ(IJK)
    AFM(3)=FEM(3)+CJ(IJK)
    AFM(4)=FEM(4)+CJ(IJK)
    AFM(5)=FEM(5)+EJ(IJK)
    AFM(6)=FEM(6)+PJ(IJK)
    AFM(7)=FEM(7)+OK(IJK)
    AFM(8)=FEM(8)+SK(IJK)
    AFM(9)=FEM(9)+RK(IJK)
    AFM(10)=FEM(10)+DK(IJK)
    AFM(11)=FEM(11)+FK(IJK)
    AFM(12)=FEM(12)+OK(IJK)
    WRITE(IT,932)IJK,(AFM(M3),M3=1,12)
120 CONTINUE
932 FORMAT(1X,I3,2X,12E9.2)
274 CONTINUE
    GO TO 977
11  WRITE(IT,133)
133 FORMAT(15X,'ERROR IN INPUT DATA,GCNE TO NEXT PROBLEM'//)
    GO TO 977
10  WRITE(IT,983)UD,N
98  FORMAT(' BANDWIDTH TCC LARGE FOR MATRIX   UD=',I3,3X,'N=',I3)
    GO TO 977
11  WRITE(IT,944)TEM,NDIM1
```

```
42900        DC 9991 J=1,12
42915  9991  T(I,J)=0.C
43020        T3=CRFT(CX**2+CZ**2)
43100        IF(ST.EC...)CO TO 7C
43200        IF(ST.EC.9C.)CO TO 71
43300  7C    CS=1.
43350        SN=C.
43500        GC TC 72
43600  71    CS=C.
43700        SN=1.
43900  6C    S=SI(S)
44030        SN=SN(S)
44130        CS=CCS(S)
44230  72    CCNTINUE
44390  6£    IF(CZ..TC.C)CO TO 6a
44400        T3=SCFT(CX**2+CY**2)
44530        T(1,1)=CX
44615        T(1,2)=CY
44700        T(1,3)=CZ
44800        T(2,1)=(-CX*CZ*SN-CY*CS)/T3
44900        T(2,2)=(-CY*CZ*SN+CX*CS)/T3
45020        T(2,3)=T3*SN
45110        T(3,1)=(-CY*CZ*CS-CX*SN)/T3
45220        T(3,2)=(-CY*CZ*CS+CX*SN)/T3
45300        T(3,3)=T3*CS
45400        GC TC 47
45500  64    T(1,1)=CX
45600        T(1,2)=CY
45700        T(1,3)=CZ
45800        T(2,1)=(-CX*CY*CS-CZ*SN)/T3
45920        T(2,2)=(-CY*CZ*CS+CX*SN)/T3
46000        T(2,3)=T3*SN
46120        T(3,1)=(-CX*CY*SN-CZ*CS)/T3
46200        T(3,2)=(-CY*CZ*SN+CX*CS)/T3
46300        T(3,3)=T3*CS
46530  4;    DC 62 K=3,9,3
46620        DC 62 I=1,3
46720        DC 62 J=1,3
46800        IK=J+K
46900        JK=J+K
47000  62    T(IK,JK)=T(I,J)
47130        RETURN
47220        END
47300  C    *******************************************
47500  C    *******************************************
47602        SUBROUTINE MEMBER(T,AE,AKT,E,OIY,OIY,OIZ,OA,G,OL,AT)
47632        DIMENSION T(12,12),AK(12,12),AKT(12,12),TS(12,12)
47730        DC 65 J=1,12
47800        DC 65 I=1,J
47900  65    AK(I,J)=0.0
48000        C1=2.*E*OIY/OL
48200        C2=2.*E*OIZ/OL
48230        C3=3.*C1/OL
48350        C4=3.*C2/OL
48400        C5=2.*C4/OL
48530        C7=G*OIY/OL
48632        C8=E*OA/OL
48730        AK(1,1)=C8
```

```
48900        AK(1,7)=-C8
49000        AK(7,7)=C8
49100        IF(T1.NE.O)CC TO 7499
49200        AK(2,2)=C6
49330        AK(3,3)=C5
49430        AK(5,4)=C7
49500        AK(5,5)=2.*C1
49600        AK(6,6)=2.*C2
49700        AK(8,8)=C6
49800        AK(6,9)=C5
49900        AK(10,10)=C7
50000        AK(11,11)=2.*C1
50120        AK(12,12)=2.*C2
50220        AK(2,6)=C4
50300        AK(2,8)=-C6
50400        AK(2,12)=C4
50600        AK(3,5)=-C3
50700        AK(3,9)=-C5
50920        AK(3,11)=-C3
51020        AK(5,9)=C3
51120        AK(5,11)=C1
51200        AK(6,8)=-C4
51300        AK(6,12)=C2
51330        AK(8,12)=-C4
51430        AK(9,11)=C3
51530  7499  DC 66 J=2,12
51600        J1=J-1
51700        DC 66 I=1,J1
51800  66    AK(J,I)=AK(I,J)
51920        DC 9994 I=1,12
52000        DC 9994 J=1,12
52130        TS(I,L)=C.C
52200        DC 9994 J=1,12
52300  9994  TS(I,I)=TS(T,I)+AK(I,J)*T(J,I)
52400        DC 9995 I=1,12
52500        DC 9995 L=1,12
52630        AKT(I,L)=C.
52770        DO 9995 J=1,12
52800  9995  AKT(I,L)=AKT(I,L)+T(J,I)*TS(J,L)
52900        RETURN
53020        END
53120  C    *******************************************
53220  C    *******************************************
53300        SUBROUTINE STCRE(T,AKT,STIF,PUE,NTIM)
53500        DIMENSION KT(12),AYT(12,12),STIF(9C30)
53530        MM=MUD-1
53630        KS=(MUD-NB)/2
53700        DC 621 L=1,12
53800        I=KT(I)
53900        DC 621 K=1,12
54000        J=KT(K)
54120        IF(I.IF.J)CO TO 621
54220        IF(I.EC.FD)M1.CR.J.EC.NDIM))GC TO 621
54300        LI=J+(I-M7)*MUD+MS
54520        IF(I.LE.MD)LI=J+(I-1)*T)/2
54632        STIF(LI)=STIF(LI)+AKT(L,K)
54730  621   CCNTINUE
54800        RETURN
54900        END
```

```fortran
C*********************************************************
C*********************************************************
          SUBROUTINE CHOLES(A,N,MM,TB,NT,IPRNT,*)
          DIMENSION A(9000)
          REAL*8 S1,T1,TMP
C
C*** A IS THE ARRAY CONTAINING THE ELEMENTS OF THE LOWER HALF BAND
C*** OF THE SYMMETRIC STRUCTURE STIFFNESS MATRIX.
C*** N=NUMBER OF EQUATIONS OR NUMBER OF UNKNOWN DISPLACEMENTS
C*** MM= HALF BANDWIDTH +DIAGONAL ELEMENT OR MOD + 1
C*** TB+1 = NO. OF RIGHT HANDSIDE VECTORS
C*** NT-- IF 1 OR LESS MATRIX IS FACTORED. IF GREATER THAN 1
C*** FORWARD AND BACKWARD SUBSTITUTION PERFORMED.
C*** S1 AND T1 ARE TEMPORARY DOUBLE PRECISION VARIABLES
C
          MOD=MM-1
          MS=MM*MM/2
          KM=MM-MS
          IF(NT-1)30,30,31
C
C*** BEGIN CHOLESKY ALGORITHM FOR FACTORING MATRIX
C
30     DO 26 J=1,N
       IF(J-MOD)1,1,2
2      IM=J-MOD
       L=I+(J-MS)*MOD+MS
       GO TO 7
1      IM=1
7      L=IM+(J-1)*J/2
       J1=J-1
       J2=J+1
       IF(J1)6,8,3
3      DO 6 I=IM,J1
       T1=A(L)
       S1=S1+T1*T1
4      L=L+1
8      T1=A(L)
       TM=T1-S1
       T1=DSQRT(TMP)
       A(L)=T1
       IF(J-N)9,20,20
9      DO 18 I=J2,N
       SUM=0.0
       IF(I-MOD)68,68,71
71     LI=IM+(I-1)*I/2
       LI=IM+(I-MS)*MOD+MS
       IF(IT-J)53,53,18
       DO 17 F=IM,J1
       IF=I+K-J
       SUM=SUM+A(LI)*A(LB)
17     LI=LI+1
18     A(LI)=(A(LI)-SUM)/A(L)
26     CONTINUE
C
C*** BEGIN FORWARD SUBSTITUTION
C
31     NP=IB+1
       YB=N+1
       DO 66 K=1,NR
       A(NB)=A(NB)/A(I)
       DO 6C I=2,M
       IF(I-MOD)21,21,22
22     IV=I-MM
       KS=IM*MOD+MS
       ME=MOD
       GO TO 27
21     IM=0
       ME=I-1
       KS=MS*I/2
27     SUM=C.C
       DO 61 J=1,MS
       JP=J+IM
       L=JP+KS
6      SUM=SUM+A(L)*A(JB)
6C     A(JB+1)=(A(JB+1)-SUM)/A(II)
65     NB=NB+N
C
C*** BEGIN BACKWARD SUBSTITUTION
C
       N2=NM+N
       DO 75 K=1,NB
       A(NP)=A(NP)/A(MM)
       DO 8C II=2,M
       I=N-II+1
       IF(I-MOD)84,84,95
95     IC=I+(I-MM)*MOD+MS
       GO TO 82
41     ID=I+(I-1)*I/2
42     IF(I-N+MM)83,83,45
4:     MS=II-1
       GO TO 76
43     MS=MOD
7C     SUM=C.0
       DO 81 J=1,MS
       JP=I+J
81     SUM=SUM+A(L)*A(JB)
82     A(JB-MOD)98,98,99
95     L=I+(JP-MM)*MOD+MS
       GO TO 82
96     L=I+(JP-1)*JR/2
85     JR=NP-N+JP
81     SUM=SUM+A(L)*A(JB)
85     JP=MS-N+I
8C     A(JB)=(A(JB)-SUP)/A(IC)
75     MR=NB+N
       RETURN
1CC    WRITE(IPRNT,9C1)
9C1    FORMAT(' STIFFNESS MATRIX IS NOT POSITIVE DEFINITE MAT THIS
      1 PROBLEM.')
       RETURN 1
       END
```

```
11800        1'SHEAR MODULUS CF RIGIDITY =',E18.8)
12000   205 FCRMAT(2Z,I2,5X,F10.2,5X,E18.P)
12500   211 FCRMAT(1H1,//5X,'ANALYTICAL METHODS PACKAGE FOR ENGINE RCO',/,
12600       15X,'THRUST BEARING FOUNDATION STIFFNESS')
12700   213 FCRMAT(//,5X,'STIFFNESS IN SHEAR DEFLECTION',//,2Y,130('*'),//,2X,
12800       1130('*'))
12900   215 FCRMAT(///,15X,'MEMB PROPERTIES',//,10X,'NO',5X,'DISTANCE FROM',5X,
13000       1'FORWARD',2X,'SECTION AREA',9X,'SECTION AREA',//,22X,'AFT END',
13100       26X,'FORWARD',1X,'AFT END',12Y,'FORWARD END',//,132('-'))
13200   216 FCRMAT(//,15X,'OUTPUT VALUES',//,10X,'NO',5X,'POSITION FROM',1CX,
13300       1'SHEAR',//,2CX,'AFT END',1CX,'DEFLECTION',//,132('-'))
13400   217 FCRMAT(//,5X,'OUTPUT VALUES',//,2X,'POSITION FROM',//,2X,
13500       1130('*'))
13600   218 FCRMAT(//,15X,'OUTPUT VALUES',//,10X,'NO',5X,'POSITION FROM',1CX,
13700       1'BENDING',//,2X,'AFT END',1CX,'DEFLECTION',//,132('-'))
13800       RETURN
13900       END
14000  C **********************************************************
14100  C **********************************************************
14200       SUBROUTINE AREVAL(FT,I,P,THETA,ADM,AEI,PM)
14300  C **********************************************************
14500  C SUBROUTINE TO FIND DISTRIBUTION OF VALUES & CALLS FUNCTION FOR AREA
14600  C OR MOMENT CALCULATION
14700  C **********************************************************
14800       COMMON/99/ZI(25),XD(25),RFP,VI(25),VD(25)
14900       DIMENSION T(50),A(50)
15000       THETA=C.0
15100       A(1)=C.C
15200       DC 4C3 I=1,NPP
15300       AED1=C.3
15400       AED2=C.C
15500       N=I*3-1
```

```
100        TRI=(XD(I)-YL(I))/8.0
200        DCR1=(XD(I)+YI(I))*0.5
300        DCR2=XD(I)
400        T1=(VD(I)-VI(I))*(*DI-XI(I))/(XD(I)-YI(I)))*VI(I)
500        T2=(VD(I)+VI(I))*0.5
600        IF(DCP1.GT.ADI)ADD1=TACM(ADI,DCP1,ADV,ADM,T1,T2,RM)
700        IF(DCR2.GT.ADI)ADD2=TCP(ADI,DCR2,ADV,ADM,T1,VD(I),RP)
800        IF(DCR3.GT.ADI)DD2=TDP(DCP1,DCR2,ADV,ADP,T2,VD(I),EM)
900        IF(VI(I).GT.ADI)ADD=TAOM(VL(I),DCR1,ADV,ADP,VI(I),T2,RM)
1000       IF(N.NE.2)N(N-1)=N(N-2)
1100       IF(XU(I).GT.ADI)GO TO 804
1200       A(N)=A(N-1)+TAOM(VL(I),DCP1,VI(N-1),VIN),1.,1.,RM)+ADD1
1300       A(N+1)=A(N)+TACM(DCP1,XD(I),VI(N),V(N+1),1.,1.,PP)+ADD2
1400       THETA=A(N+1)
1500   4C: CONTINUE
1600       GO TO 805
1700   4C4 DIS=PT-ZL(I)
1800       AED1=C.3
1900       DA=YD(I)-XL(I)
2000       T3=(VD(I)-VL(I))*(PT-XL(I))/DA*VL(I)
2100       CC3=T(N-1)
2200       CCA=(T(N+1)-Y(N-1)/D?
2300       VTD=DIS-COA*CCB
2400       IF(PT.GT.VL)DD1=TAOM(ADI,PT,ADP,ADP,T1,T3,RP)
2500       IF(XL(I).GT.ADI)AED1=TAOM(XL(I),PT,DM,ADP,VL(I),T3,EM)
2600       AREA=TACM(XL(I),PT,V(N-1),VTD,1.,1.,B')+ADD1
2700       THETA=T*PETA+AREA
2800   4C: RETURN
2900       END
3000  C **********************************************************
3100       FUNCTION TACM(X,T,V1,V2,T1,T2,EM)
3200  C **********************************************************
3300  C FUNCTION FOR AREA OR MOMENT CALCULATION DEPENDING ON FLAG
3400  C **********************************************************
3500       COMMON/92/FLAG
3600       H=T-X
3700       VAR=((V1/T1)+(V2/T2))/EM
3800       VAS=((V1/T1)+(2.0*T2/T2))/EM
3900       TAOM=H*0.5*VAR
4000       IF(FLAG.EQ.'M')TAC=((H**2)+VAS/6.0)*TACM*X
4100       RETURN
4200       END
4300  C **********************************************************
4400  C **********************************************************
4500  C **********************************************************
4600  C **********************************************************
4700  C END OF SECTION FOR THRUST BEARING STIFFNESS AND THREP SCTION STIFFNESS
4800  C TO THRUST FORCE
4900  C **********************************************************
5000       END OF PROGRAM
5100  C **********************************************************
```

8-8

DTIC